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MECHANICAL ENGINEERING

INCLUDING THE ENGINEERING INDEX

Annual Meeting Papers

This Annual Meeting advance issue contains twenty-three complete papers and synopses of twenty papers (to be printed in pamphlet form) that are to be presented at the Annual Meeting of the A.S.M.E. in New York, November 30 through December 4, 1925. The papers of which synopses are given in this issue may be secured by filling out the form just preceding the advertising section.

Bring this copy to the Meeting as papers appearing in it will not be available for free distribution. However, additional copies may be purchased if you do not care to be burdened with it.

MID-NOVEMBER 1925

THE MONTHLY JOURNAL PUBLISHED BY THE AMERICAN SOCIETY OF MECHANICAL ENGINEERS

Mechanical Engineering

The Monthly Journal Published by

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Volume 47 November, Part 2, 1925 Number 11a

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Purpose of this Issue

THE basic policy which guides the planning and conduct of A.S.M.E. meetings encourages the widespread participation in their proceedings by all members of the Society. To carry out this policy the Committee on Meetings and Program requires that manuscripts be ready well in advance of the meeting so that they may be printed. The Committee on Publications coöperates in this method of disseminating the papers widely before the meeting so that every member may have an opportunity to study the papers carefully and present well-considered discussions. Papers that are printed in advance are presented only in abstract at the Meeting, giving the major portion of the time of the session over to discussion.

This special issue contains twenty-three complete papers and synopses of nineteen others which are to appear in pamphlet form and may be secured by filling out the form just preceding the advertising pages of this issue.

This issue should be brought to the Meeting, as copies of the papers appearing in it will not be available for free distribution. However, additional copies of it may be purchased at the Meeting.

Outstanding Events of the Meeting

THE tentative program of the Forty-sixth A.S.M.E. Annual Meeting on the opposite page reveals a wealth of valuable technical material produced by the painstaking efforts of members of the Society under the guidance of the Professional Divisions. The excellence of this program attests anew the service that A.S.M.E. meetings are rendering as great clearing-houses of information and inspiration.

An important innovation in the technical program of the Society for the coming meeting is the addition of the Henry Robinson Towne and the Robert Henry Thurston Lectures. These lectures are to deal with problems in the border zones between engineering and economics, and engineering and science, respectively, and the lecturers for 1925 are prominent leaders of thought in these two fields. Hon. Herbert Hoover, Secretary of Commerce, will deliver the Towne Lecture, and Dr. Zay Jeffries, a noted consulting metallurgical engineer, the Thurston Lecture.

An elaborate program of entertainments and excursions for both ladies and men is well under way. The feature of this part of the program will be the Dinner on Wednesday evening, December 2, at the Hotel Astor, which will follow the procedure inaugurated last year which provided such thorough enjoyment for those who attended.

The American Society of Refrigerating Engineers will meet in New York during the week of the A.S.M.E. Meeting, with headquarters at the Hotel Astor. A joint session of the A.S.M.E. and the A.S.R.E. is scheduled for Tuesday afternoon, December 1.

The Taylor Society will meet in the Engineering Societies Building on Friday and Saturday of the A.S.M.E. Meeting week. The Taylor Society is coöperating with the A.S.M.E. Management Division in the conduct of two sessions on Thursday, December 3.

As usual, the National Exposition of Power and Mechanical Engineering will be held in the Grand Central Palace during the week of the Meeting. The coming exposition will entirely fill three floors of the Grand Central Palace with four hundred interesting exhibits of all types of mechanical-engineering equipment.

MECHANICAL ENGINEERING

Volume 47 Mid-November, 1925 No. 11a

Tentative A.S.M.E. Annual Meeting Program

New York, November 30 to December 4, 1925

Monday, November 30

Morning: 9:30 A.M.
Council Meeting.
Local Sections Delegates' Conference.

Noon:
Luncheon of Council and Local Sections Delegates.

Afternoon: 2:00 P.M.
Council Meeting.
Local Sections Delegates' Conference.
Main Committee on Power Test Codes.

Evening:
Open House.

Tuesday Morning, December 1, 9:30 A.M.

Oil and Gas Power

(Under auspices of Oil and Gas Power Division)

Centrifugal Compressors for Diesel Engines, SANFORD A. MOSS.
Auxiliaries for Motor Vessels, J. W. MORTON and A. B. NEWELL.
Internal-Combustion-Engine Transmissions, HERMANN LEMP.
Gas Engines of Maryland Plant of Bethlehem Steel Co., A. A. RAYMOND. (By title)

Machine Shop Practice (I)

(Under auspices of Machine Shop Practice Division)

Principles and Advantages of Optical Methods for Measuring Machine Parts, HENRY F. KURTZ.
The Tension Ratio and Transmissive Power of Belts, C. A. NORMAN.

Wood Industries Session

(Under auspices of Wood Industries Division)

The Electric Molder, A. JENSEN, JR.
Scientifically Developed Shipping Containers, C. M. BONNELL, JR.
High-Speed Induction Motors and Frequency Changers, CHAS. FAIR.
Recent Advances in Methods of Glue Evaluation, W. L. JONES.
Spark Arresters and Forest Fires, J. S. MATHEWSON.

Public Hearing

Power Test Codes on Steam Turbines.

Tuesday Afternoon, 2:00 P.M.

Industrial Power

(Under auspices of Power Division)

The Value of Higher Steam Pressures in the Industrial Plant, WM. F. RYAN.
The Supply of Industrial Power, W. H. LARKIN, JR.

Railroad

(Under auspices of Railroad Division)

Locomotive Boilers, author to be announced.
Car Design, author to be announced.

Centrifugal Compressors

(Joint Session with American Society of Refrigerating Engineers)

Centrifugal Compression as Applied to Refrigeration, W. H. CARRIER.
The Heat-Balance Method of Testing Centrifugal Compressors, M. G. ROBINSON.

Calculation Methods

Graphical Methods of Calculation, H. L. SEWARD.
The Making of Special Slide Rules, G. W. GREENWOOD.

Lecture, 4:30 P.M.

Henry Robinson Towne Lecture on Engineering and Economics by HON. HERBERT HOOVER, Secretary of Commerce.

Tuesday Evening

Award of Honorary Memberships to HON. HERBERT HOOVER and WORCESTER R. WARNER, Past-President, A.S.M.E.

Presidential Address and Reception.

Wednesday Morning, December 2, 9:30 A.M.

Industrial Furnaces

(Under auspices of Fuels Division)

Industrial-Furnace Efficiency, V. J. AZBE.
Fuels and Furnaces for Industrial Heating, W. TRINKS.

Materials Handling

(Under auspices of Materials Handling Division and A.S.M.E. Safety Committee)

Materials-Handling Problems and Their Solution, FRANK D. CAMPBELL.
Safety in Materials Handling, D. S. BEYER.

Machine Shop Practice (II)

(Under auspices of Machine Shop Practice Division)

Some Comparative Wear Experiments on Cast-Iron Gear Teeth, G. H. MARX, L. E. CUTTER and B. M. GREEN.
Normal Pitch, The Index of Gear Performance, G. M. EATON.
Question Marks in Machine Design, F. E. CARDULLO.

Springs

(Under auspices of Special Research Committee on Metal Springs)

Phosphor-Bronze Helical Springs from the Standpoint of Precision Instruments, W. G. BROMBACHER.
The Manufacturing of Commercial Steel Helical Springs, F. H. BROWN.
Characteristics of Weighing Springs, J. W. ROCKEFELLER, JR.
Springs for Electrical Measuring Instruments, B. W. ST. CLAIR.
Formulas for the Design of Helical Springs of Square or Rectangular Steel, C. T. EDGERTON.
An Outline for the Application of Fatigue and Elastic Results to Metal-Spring Design, T. MCLEAN JASPER.
The Ring Spring, O. R. WIKANDER.

Wednesday Noon

Luncheon of Council and Student Branch Delegates

Wednesday Afternoon

General, 2:00 P.M.

Annual Report of Council.
Award of Charles T. Main, Junior and Student Prizes.
Report of A.S.M.E. Investigation for S.P.E.E. on mechanical engineering education.
Progress Reports of A.S.M.E. Professional Divisions.

Education and Training for Industries of Non-College Type, 3:00 P.M.

(Under auspices of Committee on Education and Training for the Industries)

Education for the Industries of Non-College Type in Europe, WM. E. WICKENDEN.
The Apprenticeship Movement and Its Relation to the Building Trades, FRANCIS MAHONEY.

Steam Tables Research, 3:00 P.M.

Student Branch Conference, 3:00 P.M.

Ladies' Tea and Reception, 3:30 P.M.

Wednesday Evening

Annual Dinner at Hotel Astor

Thursday Morning, December 3, 9:30 A.M.

Steam Power

(Under auspices of Power Division)

Radiation in Boiler Furnaces, B. N. BROIDO.
Recent Developments at Colfax Station, C. W. E. CLARKE.
Steam Bleeding and Turbine Performance, C. D. ZIMMERMAN.
The Development of a Unit Pulverizer, R. SANFORD RILEY and O. CRAIG.

Management

(Under auspices of Management Division and Taylor Society)

The Influence of Plant Design on Plant Efficiency, HAROLD T. MOORE.
Production Control in the Newsprint Industry, GEO. D. BEARCE.
Carbon Dioxide as an Index of Fatigue, W. N. POLAKOV.

Aeronautics

(Under auspices of Aeronautic Division)

Technical Progress at McCook Field, LT. E. E. ALDRIN.
The Airship and Its Place in Commerce, LT-COL. H. H. BLEE.
Maintenance and Depreciation of Airplanes and Motors, LT. E. W. DICHMAN.

Textiles

(Under auspices of Textile Division)

Application of Individual Motors to Carding Machines, WM. A. MAYOR.
The Full Automatic Loom, CHAS. F. MERRILL.

Thursday Afternoon, 2:00 P.M.

Power-Plant Materials

(Under auspices of Power Division)

Steel Castings for Use in High-Pressure Steam Lines, A. E. WHITE.
The Quality of Bolts for Use in Power-Plant Construction, W. P. WOOD.
Furnace Refractories, E. B. POWELL.

Industrial Psychology

(Under auspices of Management Division and Taylor Society)

Present State of Industrial Psychology, LILLIAN M. GILBRETH.
Psychology in Organization of Prison Industries, E. A. DOLL.

Lubrication

(Under auspices of Special Research Committee on Lubrication)

Graphical Study of Journal Lubrication, Part 3, H. A. S. HOWARTH.
Charts for Studying the Oil Film in Bearings, G. B. KARELITZ.

Design

Torsional-Stress Concentrations in Shafts of Circular Cross Section and Variable Diameter, L. S. JACOBSEN.
Tangential Vibration of Steam-Turbine Buckets, W. CAMPBELL and W. C. HECKMAN.
Vibration Phenomena of a Loaded Unbalanced Shaft while Passing through the Critical Speed, A. L. KIMBALL, JR., and E. H. HULL.

Lecture, 4:30 P.M.

Robert Henry Thurston Lecture on Engineering and Science in the Metal Industry, by DR. ZAY JEFFRIES, Metallurgical Engineer.

Thursday Evening

National Defense

(Under auspices of National Defense Division)

Speakers to be announced

Friday, December 4

Council Meeting—Committee Meetings—Excursions

The Airship and Its Place in Commerce

A General Survey of the Development of the Airship and a Discussion of Its Commercial Possibilities

By H. H. BLEE,[1] SANTA ANA, CAL.

THE aspiration to fly has prompted man since the remotest times to ponder upon the possibilities of human flight. When he first undertook the study of this problem, he naturally turned his attention to the two things he observed in nature which seemed to possess an inherent power to rise into the air and to remain aloft—birds and smoke. These early studies soon led to two distinct schools of aeronautical thought which, after years of experimentation, have developed the heavier-than-air and lighter-than-air types of aircraft of the present day. Thus it is that the origin of the airplane and airship, or dirigible, which are seen in flight together in Fig. 1, can be traced directly to the practical development of the old bird-flight and smoke-flight theories.

General Characteristics of the Airship

The airship is essentially an elongated balloon of streamline form fitted with both horizontal and vertical stabilizing and control surfaces and with one or more cars for the accommodation of power plants, operating crew, passengers, and express. By releasing ballast (usually in the form of water) or gas, as conditions may require, the ship may be caused to ascend or descend vertically under the influence of the static forces, or it may be brought to rest in a condition of static equilibrium at any desired altitude within its altitude range. On the other hand, by the use of the motors and horizontal control surfaces an airship may be driven along a horizontal path or upward or downward along an inclined path regardless of whether it is, statically, somewhat heavy or light or in equilibrium. When flying at full speed, an airship can develop a maximum dynamic lifting force equal to approximately 10 per cent of the gross static lift of the ship. This depends, of course, upon the power, speed, and design of the ship and the altitude at which it is flying. The dynamic lift is a maximum when the axis of the ship is inclined at an angle of about 10 deg. to the horizontal, and may be either positive or negative, depending upon whether the ship is inclined upward or downward. This force is taken advantage of in "taking off" and flying a ship which is statically heavy, in making changes in altitude without changing the static condition of the ship, as indicated above, and in landing a ship that has become statically light due to the consumption of fuel, superheating of the gas, or the release of too much ballast.

Fig. 1 U. S. Airship "Shenandoah" and a DH-4 Army Airplane in Flight Near Scott Field, Ill.

Historical Development

The first motor-driven airship was built in the year 1852 by a Frenchman named Henri Giffard, the inventor of the steam injector. This was a small ship of the non-rigid type, with a displacement of 88,000 cu. ft., and was driven by a steam engine developing from 3 to 5 hp., which gave it a speed in still air of about 5 miles per hour. Airships did not become a practical proposition, however, until after the advent of the light internal-combustion engine.

The first attempt at rigid-airship construction was made in 1897 by an Austrian named Schwartz, who built a 130,000-cu. ft. ship, using a framework of aluminum tubing covered with sheet aluminum, and fitted it with a small gasoline motor belted to two propellers. This ship met with disaster on its first flight. The driving belts came off their pulleys, leaving the ship adrift in the air without power, and it was completely wrecked in landing.

In 1898 Santos-Dumont, a Brazilian, began a series of experiments in airship construction in Paris. He built a number of non-rigid ships powered with small motor-car engines. His last ship, which was brought out in 1904, had a displacement of some 22,000 cu. ft., was equipped with a 16-hp. motor and made a speed of about 16 miles per hour.

In 1899 the French engineer Juillot began the construction of a series of small semi-rigid airships. These were the first ships of this type and were quite successful.

The first airship to be constructed in the United States is shown in Fig. 3. This ship was designed, built, and flown in the year 1900 by A. Leo Stevens. The envelope had a capacity of 34,000 cu. ft., was made of varnished Japanese silk, and was covered with a diamond-mesh balloon net which supported the car. The ship was fitted with a 7½-hp. engine and had a flying speed of 14 miles per hour. Horizontal control was obtained by means of a rudder mounted on the rear end of the car, while vertical control was effected by the operator shifting his weight forward or aft in the car, thereby inclining the axis of the ship up or down as might be desired. Mr. Stevens continued his experimental work over several years and sold a number of these non-rigid ships for exhibition purposes.

In 1898 Count Zeppelin, of Germany, believing that the success of large airships would be found in a rigid type of construction, began his studies and experimental work on ships of that type. His first experimental ship, the forerunner of the rigid airships of today, was brought out in 1900. This ship was 420 ft. long, 38 ft. in diameter, and had a gas capacity of 400,000 cu. ft. It was powered with two 16-hp. engines and made a speed of about 20 m.p.h.

In modern airship construction, France leads in the development of non-rigid ships; Italy has made the greatest strides in ships of the semi-rigid type; and Germany stands first in the development of the rigid airship. However, with the completion of the commercial airship ZR-3 (now the U. S. Airship *Los Angeles*), airship construction in Germany has been brought to a standstill.

[1] Lieutenant-Colonel, Air Servi e, Airship Pilot, Mem. A.S.M.E.

Contributed by the Aeronautics Division for presentation at the Annual Meeting, New York, November 30 to December 4, 1925, of The American Society of Mechanical Engineers. All papers are subject to revision. Photos, except for Figs. 3 and 7 to 11, by U. S. Army Air Service; photos for Figs. 7 to 11 by U. S. Navy.

Be sure to bring this copy with you to the Annual Meeting

The progress made in airship construction since the building of the first ship, in 1852, is well shown by the long non-stop flights made by some of the German ships during the war (notably the flight of the L-59 from Bulgaria to East Africa and return—a distance of over 5000 miles made at an average speed of 57 miles per hour), and by subsequent long flights made by airships under peace conditions. The first transatlantic flight was made by the British Airship R-34 in her flight from Edinburgh to New York and return in July, 1919, covering the distance of 3559 miles to New York in 108 hr. 12 min. In October, 1924, the U. S.

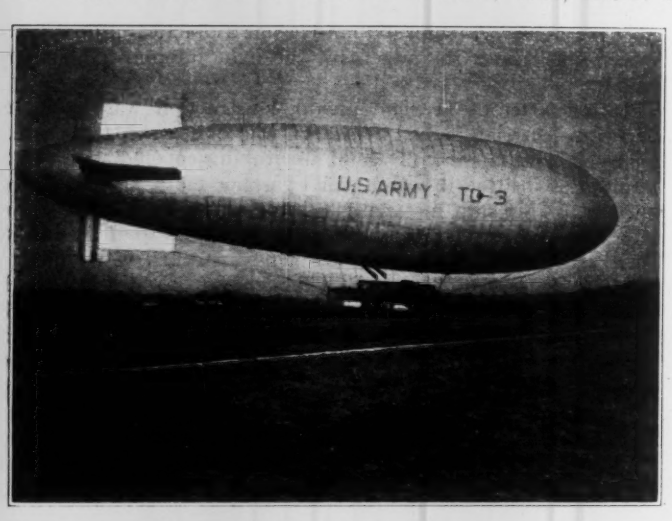

FIG. 2 A TYPICAL NON-RIGID AIRSHIP

Airship *Shenandoah* completed a 9000-mile round-trip flight from Lakehurst, N. J., to Seattle, Wash., via Fort Worth and San Diego. During the same month the ZR-3 made her non-stop flight from Friedrichshafen, Germany, to Lakehurst, N. J., a distance of 5006 miles, in 81 hr. 17 min.

NON-RIGID AIRSHIPS

Airships are divided into three distinct types or classes: namely, non-rigid, semi-rigid, and rigid. This classification is based upon the manner of maintaining the form of the gas container and the methods used in supporting from it the load carried by the ship.

In ships of the non-rigid class the form of the gas container, or envelope, is maintained entirely by internal pressure—usually about $1^1/_4$ in. of water—and the load is carried by a system of suspensions attached directly to the envelope. In order to compensate for the expansion and contraction of the gas the container is fitted with internal flexible air chambers of balloon fabric, called "ballonets." The total ballonet capacity is generally from one-quarter to one-third of the envelope capacity. There are ordinarily two of these air chambers, one located forward and the other aft, to permit trimming the ship by shifting the center of buoyancy. Air is forced into these chambers either by an independent blower or by means of metal scoops which may be lowered into the air streams from the propellers.

The following are the principal characteristics of the non-rigid Army Airship TC-3 shown in Fig. 2: envelope capacity, 210,630 cu. ft.; length, 195.8 ft.; maximum diameter, 44.5 ft.; fineness ratio, 4.40; ballonet capacity, 53,720 cu. ft. (25.5 per cent of envelope volume); ceiling, 10,000 ft.; engines, two Wright Model I 150-hp. (mounted on outriggers); inflation gas, helium; total lift, 11,585 lb.; weight of ship, 7468 lb.; useful load, 4117 lb. (35.5 per cent of total lift); number of seats, 8; full speed, 60 m.p.h.; cruising speed, 45 m.p.h.; cruising radius, 1500 miles.

The practical upper limit of capacity of non-rigid airships, as determined by structural considerations, is about 500,000 cu. ft. Based on present-day practice, ships of this type have a disposable (useful) lift ranging from 575 lb. to $4^1/_2$ tons, a ceiling of from 7500 to 12,000 ft., a cruising radius of from 500 to 1500 miles, and a maximum speed of from 40 to 60 miles per hour. The ratio of the useful lift to the maximum gross lift in non-rigid ships ranges from about 25 per cent to 37.5 per cent.

SEMI-RIGID AIRSHIPS

In semi-rigid airships the form of the gas container is maintained partly by internal gas-ballonet pressure and partly by a light, rigid keel structure (generally of triangular cross-section) which extends along the under side of the ship from bow to stern, distributing the load of the sustained weights over the entire length of the envelope. The forward end of the keel supports a light framework for maintaining the proper form of the nose of the ship, while the after end supports the directional rudder and in some cases the entire system of stabilizing and control surfaces. With this type of construction the envelope is subjected to less strain than in the non-rigid type, the ship is operated with a much lower internal pressure, and a lighter fabric is used in proportion to the size of the ship. Also, it is possible to divide the gas container into a number of compartments by transverse bulkheads of fabric. This arrangement prevents building up excessive gas pressures in the upper end of the envelope in case the ship is inclined at a steep angle to the horizontal. Furthermore, in case of a tear or break in the fabric the loss of gas is generally limited to a single compartment, the rigidity of the keel making it possible to bring the ship in under control, or at least to land it as a free balloon.

The fabric used in the construction of ships of the non-rigid and semi-rigid types is a strong rubberized cotton fabric of low permeability, one of the inner plies of which is laid on the bias to resist tearing. The average weight of three-ply airship fabric is about 14 ounces per square yard, the tensile strength is approximately 130 lb. per lineal inch, and the permeability to hydrogen is from

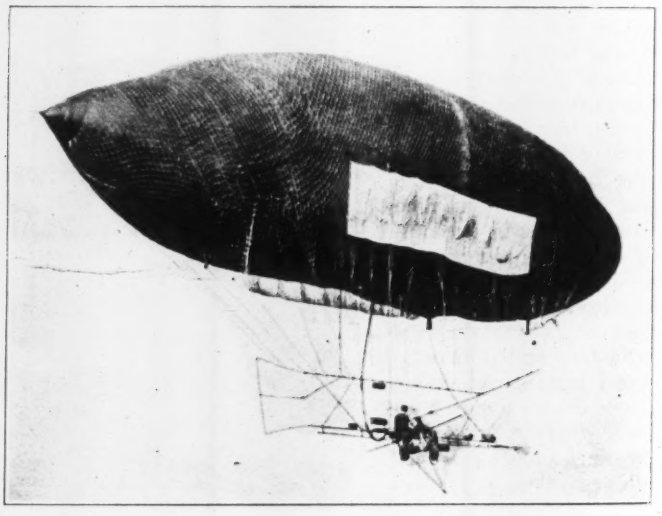

FIG. 3 THE FIRST AMERICAN AIRSHIP, BUILT BY A. LEO STEVENS IN THE YEAR 1900

(Gas displacement, 34,000 cu. ft.; horsepower, $7^1/_2$; maximum speed, 14 m.p.h. Photo courtesy of Mr. Stevens.)

10 to 15 liters per square meter per 24 hours. The film of rubber on the exterior surface of the fabric contains aluminum powder which serves to protect the rubber from the deteriorating action of the sunlight and, at the same time, to prevent sudden changes in the temperature of the contained gas by reflecting the rays striking the envelope. All seams are cemented with rubber cement and double-stitched and covered inside and out with rubberized tape cemented in place.

The limiting practical gas displacement for ships of the semi-rigid type seems to be about 1,250,000 cu. ft., although some claim a higher figure. The ratio of useful lift to maximum gross lift ranges from 35 to 50 per cent. Modern semi-rigids, depending upon their size, have a useful lift of from 1000 lb. to 20 tons, a ceiling of from 10,000 to 15,000 ft., a cruising radius of from 900 to 2500 miles, and a maximum speed of from 40 to 70 miles per hour.

RIGID AIRSHIPS

The rigid airship maintains its form by means of a light, stiff metal frame, or hull structure, regardless of the gas pressure. The control and passenger cars, power gondolas, and stabilizing and

control surfaces are attached directly to this rigid structure, which takes all shearing and bending stresses to which the ship may be subjected. The gas is contained in a number (generally from 12 to 24) of separate drum-shaped cells, the function of which is to give lifting power only.

The hull is constructed of longitudinal girders extending from bow to stern with a series of transverse frames spaced at regular intervals. Diagonal wire bracing is used between the longitudinal and transverse members and on the inside of every alternate frame, making a gas compartment for every two frames. An internal keel construction of triangular section, illustrated in Fig. 8, extends throughout the length of the hull. This keel gives added strength to the structure, distributes the weight of the concentrated loads, and at the same time affords a continuous passage from one end of the ship to the other, for the use of the operating crew, and space for the accommodation of fuel and oil tanks, ballast bags, supplies, living quarters for the crew, and compartments for mail and express. The structural members surrounding each gas compartment are interlaced on their inner surfaces with a cord netting, which serves to distribute the load over the upper surfaces of the gas cells and to confine the cells to their proper limits. The exterior of the hull is covered with a light, tightly stretched cotton fabric laced securely in place, painted with cellulose dope containing aluminum powder, and varnished.

The girders used in rigid-airship construction are for the most

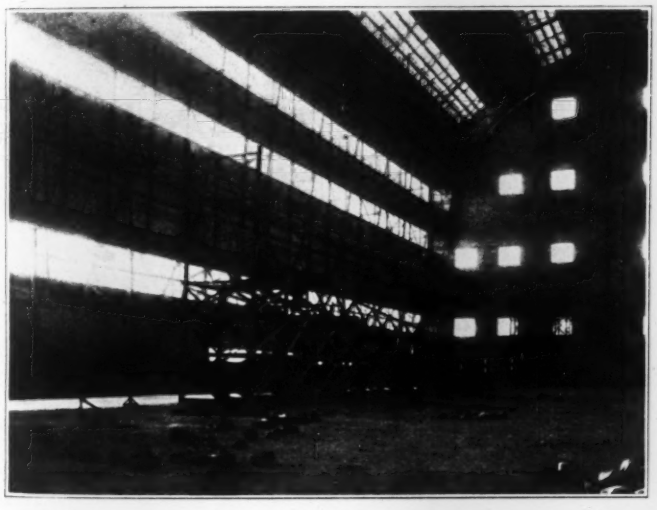

FIG. 4 KEEL STRUCTURE OF THE 750,000-CU. FT. SEMI-RIGID AIRSHIP RS-1
(Note arrangement of fuel tanks and ballast bags.)

part of the lattice type, consisting of from two to four longitudinal members of channel or U-section connected by transverse or diagonal lattice bars as shown in detail in Fig. 11. Girders of triangular cross-section with three longitudinal members predominate. The lattice bars are generally stamped from sheet metal and are either corrugated lengthwise with flat ends or are of channel section. Thus they are easily crossed and riveted at the center with the smooth faces together, one bar being riveted to the inside and the other to the outside of the longitudinal members.

The metal used in modern airship construction is almost invariably duralumin, an aluminum-copper-magnesium alloy containing up to approximately 4 per cent of copper and 1 per cent of magnesium. It is an ideal metal for this purpose in that it combines strength, hardness, and low specific gravity and does not rust. The hardness is imparted by the copper, the magnesium serving to increase the hardening effect. The successful working and use of duralumin depend almost entirely on proper heat treatment. When annealed by heating to a temperature of about 365 deg. cent. and quenching, it becomes plastic and may be forged, stamped, drawn, or rolled. When heated to a temperature of about 500 deg. cent. and quenched, it becomes hard and has approximately the same strength as structural steel. Its weight is about

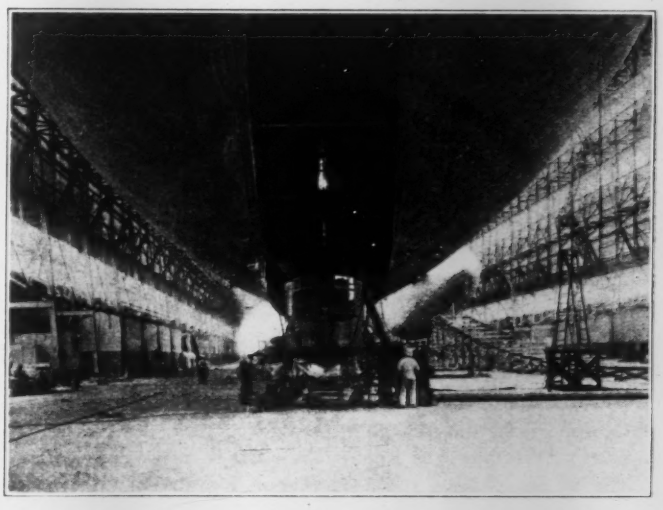

FIG. 5 PROGRESS PICTURE OF THE RS-1, SHOWING FRONT OF CONTROL CAR, A PORTION OF THE ENVELOPE, AND THE INTERIOR OF THE KEEL. TAKEN BEFORE LACING ON THE UNDER COVER

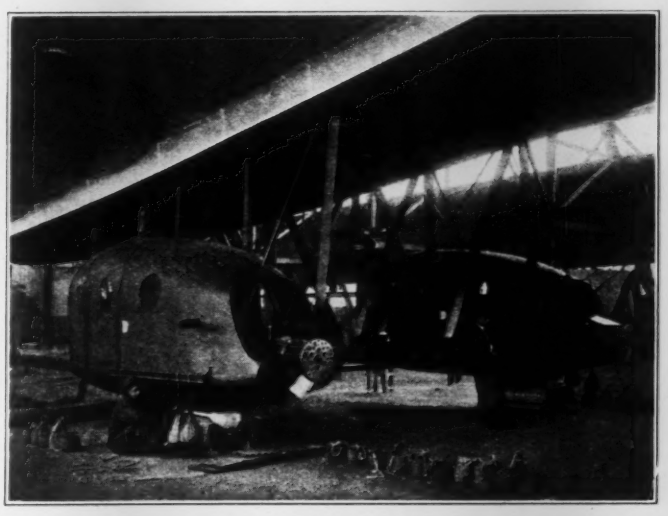

FIG. 6 THREE-QUARTER REAR VIEW OF THE POWER GONDOLAS OF THE RS-1. TAKEN BEFORE PLACING THE PROPELLERS

TABLE 1 CHARACTERISTICS OF TYPICAL MODERN RIGID AIRSHIPS

Name or designation No.	Where built	Year completed	Air displacement, cu. ft.	Gas displacement, cu. ft.	Length, ft.	Max. diam., ft.	Fineness ratio	No. of gas cells.	Inflation gas	Unit lift, lb. per cu. ft. (Hydrogen, std. lift)	Max. gross static lift, tons	Wt. of ship, tons	Useful lift, tons	Lift ratio, per cent	No. of engines	Name or make of engines	Hp. per engine	Total hp.	Max. speed, m.p.h.	Cruising hp., approx.	Cruising speed, m.p.h., approx.
R-34	England	1919	.....	2,023,450	645	78.8	8.18	19	H	0.068	68.8	32 approx.	36.8 approx.	53.5 approx.	5	Sunbeam "Maori"	270	1350	60	800	50
Shenandoah ZR-1	U. S. A.	1923	2,290,000	2,148,070	680.2	78.8	8.64	20	H	0.068	73	39	34	46.6	5[1]	Packard	300	1500	62.5	900	52.5
Los Angeles ZR-3	Germany	1924	2,764,000	2,624,583	658.3	90.7	7.25	14	H	0.068	89.2	45.0	44.2	49.5	5	Maybach	400	2000	75	1200	63

NOTE.—For the sake of comparison, all lift figures have been given on the basis of using hydrogen as the lifting medium.

[1] The *Shenandoah* was originally equipped with six 300-hp. engines; the forward engine, which was installed in the after compartment of the control car, was later removed and the compartment converted into a radio room.

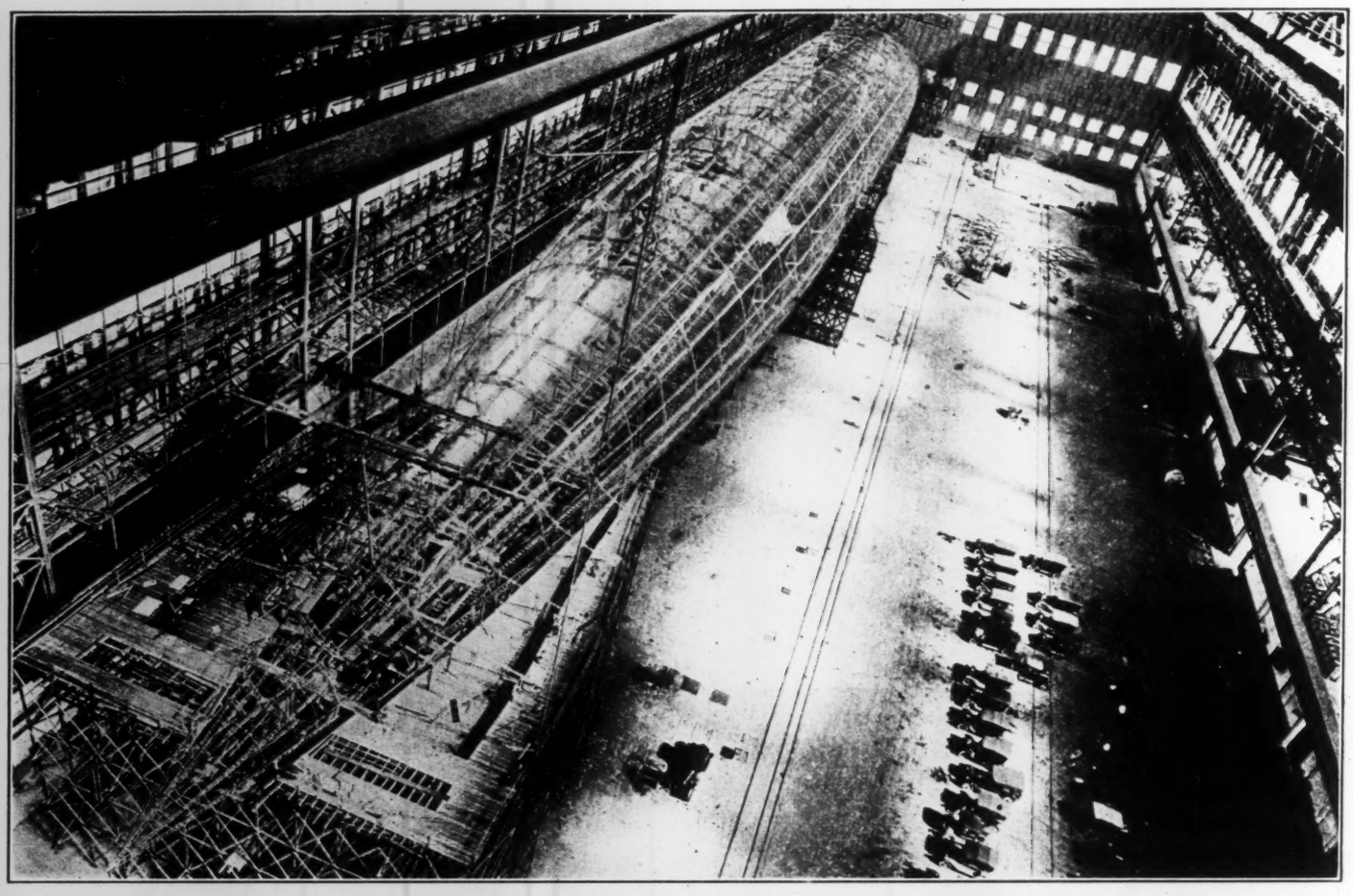

FIG. 7 PROGRESS PICTURE SHOWING HULL STRUCTURE OF THE U. S. AIRSHIP "SHENANDOAH"

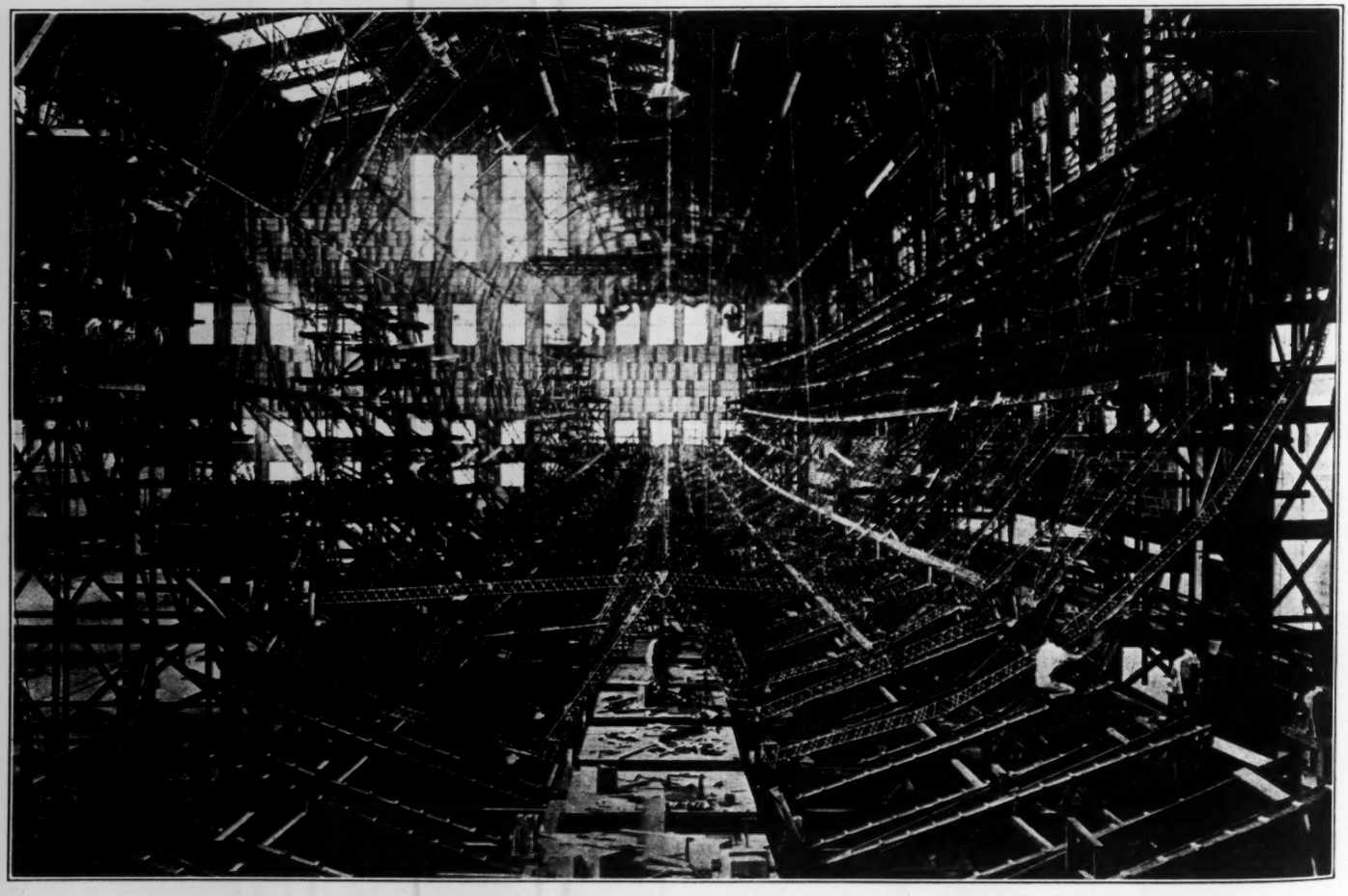

FIG. 8 INTERIOR VIEW OF HULL STRUCTURE OF THE "SHENANDOAH" SHOWING THE INTERNAL KEEL CONSTRUCTION

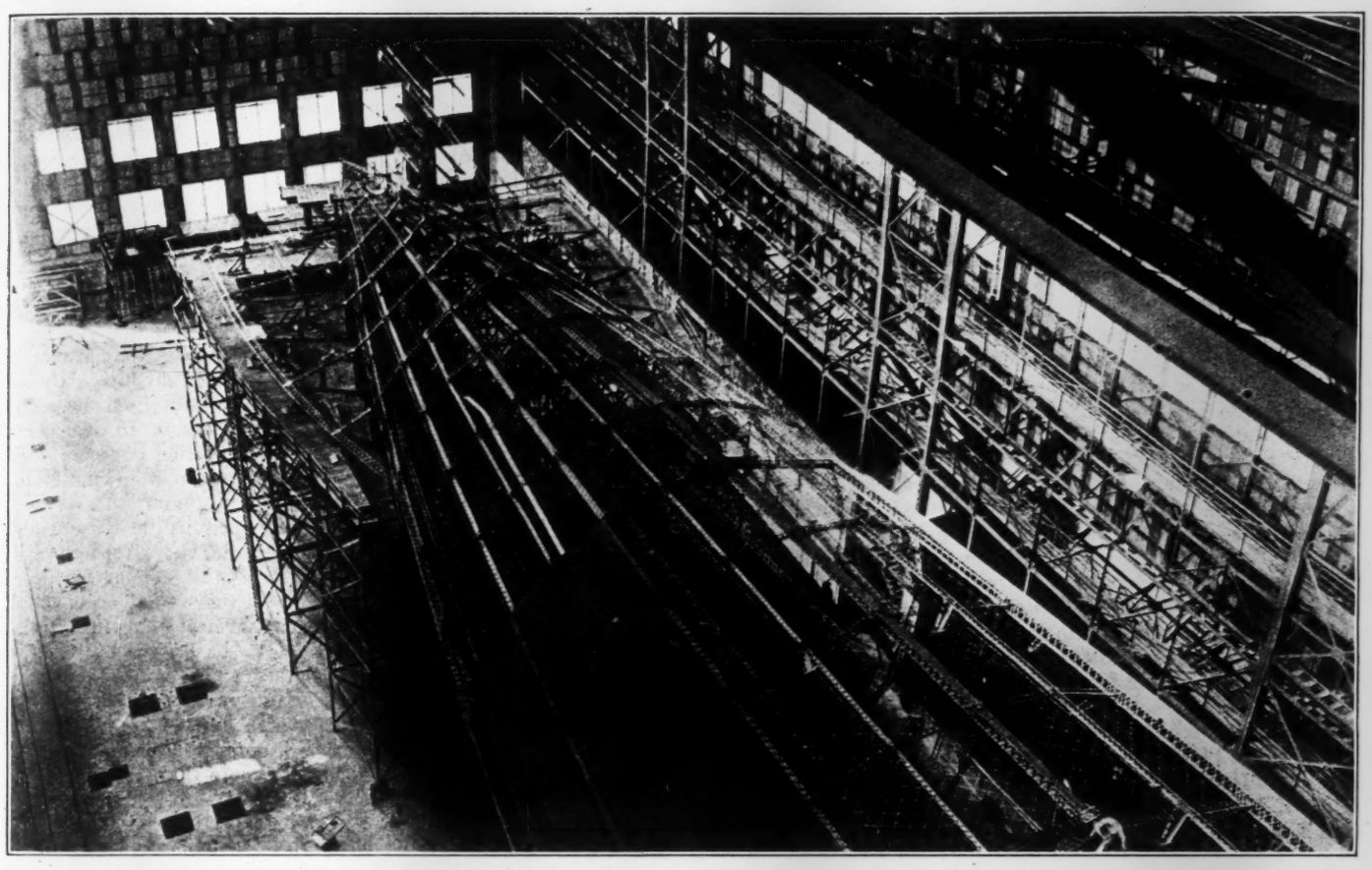

FIG. 9 AFTER PORTION OF THE "SHENANDOAH," SHOWING THE STERN CONSTRUCTION, FRAMEWORK OF STABILIZING SURFACES, CORD NETTING SURROUNDING GAS CELLS, AND A SECTION OF OUTER COVER

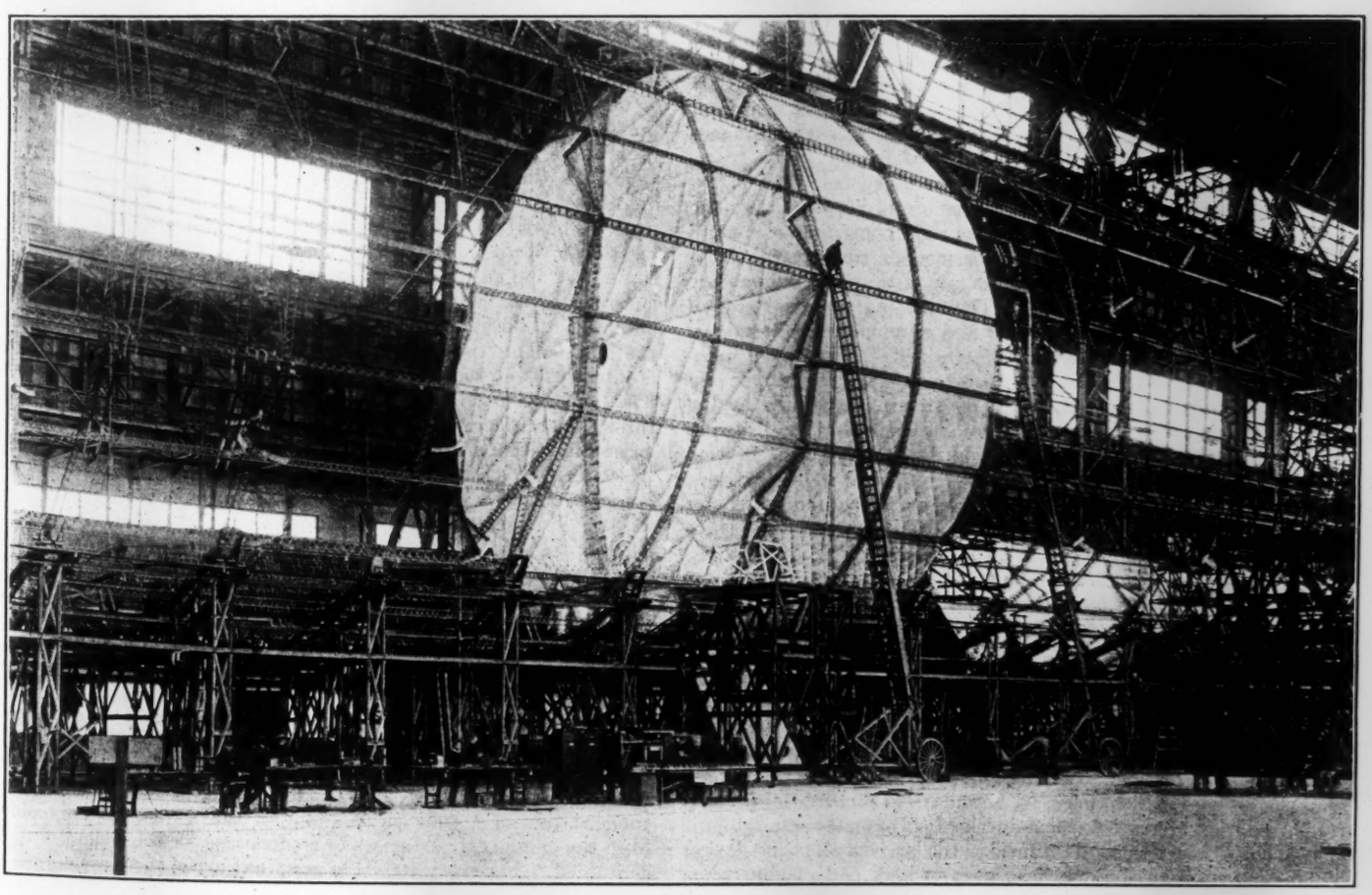

FIG. 10 ONE OF THE MID-SECTION GAS CELLS OF THE "SHENANDOAH"

(Note the system of interior wire bracing and the king-post trusses of the alternate transverse frames; also the gas trunk extending up through the center of the ship along the face of the cell.)

one-third that of steel. The physical properties of duralumin are as follows:

Specific gravity	2.80–2.85
Weight	172.8 lb. per cu. ft.
Melting point	650 deg. cent. (1200 deg. fahr.)
Coefficient of linear expansion	0.0000226 per deg. cent. 0.0000126 per deg. fahr.
Tensile strength	52,000 lb. per sq. in.
Yield point in tension	32,000 lb. per sq. in.
Compressive strength	44,000 lb. per sq. in.
Modulus of elasticity	9,400,000 lb. per sq. in.

Since the gas cells of rigid airships are not called upon to carry any direct loading, as in the case of the non-rigid and semi-rigid ships, they may be constructed of much lighter material; nor is it necessary that they be fully inflated. In fact, it is customary to inflate the cells to about 85 to 90 per cent fullness at the ground, thereby leaving room for expansion of the gas with increase in altitude. An automatic safety valve is fitted near the bottom of each cell to prevent the internal pressure exceeding a predetermined amount. These automatic valves discharge into vertical gas trunks extending to cowled openings in the top of the ship. In addition to the automatic safety valves the cells are equipped with maneuvering valves (operated from the control car) which are located at the top and discharge into these same gas trunks. The cells are made of a single-ply cotton fabric weighing from 1.5 to 2 ounces per square yard. The inner surface of this fabric is rubberized, lined with specially prepared goldbeaters' skins (the outer covering of the blind gut of cattle), and varnished. The finished fabric has a weight of from 4 to 4.5 ounces per square yard, a tensile strength of from 45 to 50 pounds per lineal inch, and a permeability of about 2 liters per square meter per 24 hours. It is understood that a new process has been developed by Mr. Norman Meadowcroft in which the goldbeaters' skins are applied to plain non-rubberized cloth by means of a gelatinous compound, thereby giving a finished fabric of longer life at a lower cost.

The rigid type of airship construction affords convenient attachment for multiple power plants at such points as to give proper distribution of load and at the same time avoid slipstream interference. One power car, or gondola, is generally located well aft on the center line of the ship, with one or more pairs of wing cars located at points forward of this position and at such distances out from the center line as to prevent interference.

For airships having a gas displacement much below 1,000,000 cu. ft., the semi-rigid type gives a higher useful lift ratio than the rigid. In the larger sizes, however, the rigid construction is more efficient. The maximum lift ratio for modern rigids varies from 45 to 60 per cent, depending upon the size of the ship, refinements of design, and factors of safety. The limiting practical size of rigid airships, as fixed by structural and economic considerations, has not yet been approached or determined. The Italian aeronautical engineer G. A. Crocco, in a paper on Structural and Economic Limits to the Dimensions of Airships (reprinted by the National Advisory Committee for Aeronautics in Technical Memorandum No. 274), draws the conclusion that the practical limiting size may well reach a maximum gross lift of 1000 metric tons (2204.6 lb. per metric ton), corresponding to a maximum diameter of 60 meters (197 ft.). Present-day rigid airships, including ships now under construction, have a disposable lift of from 15 to 80 tons, a maximum speed of from 60 to 80 miles per hour, a cruising radius of from 3000 to 6000 miles, and a ceiling up to 20,000 ft.

Comparison of Rigid with Non-Rigid and Semi-Rigid Types

The rigid airship, with its stiff metal hull structure taking all shearing and bending stresses and making possible a more perfect streamline form and a more efficient arrangement of the multiple power plants, and with its large number of independent gas cells, has a number of distinct advantages over ships of the non-rigid and semi-rigid types which may be summarized briefly as follows: (1) No internal pressure to be maintained; (2) greater aerodynamic efficiency; (3) greater useful lift ratio in the large sizes; (4) no danger of building up excessive gas pressures in high end of ship when axis is inclined to horizontal; (5) access during flight to any important part of structure for inspection or repair; (6) longer useful life (due to the comparatively rapid deterioration of the rubberized fabrics used in the construction of the non-rigid and semi-rigid types); (7) greater airworthiness, safety, and reliability; (8) may be built in much larger sizes and hence with greater load-carrying capacity, greater cruising radius, and more commodious and comfortable accomodations for passengers. On the other hand the non-rigid and semi-rigid types are of simpler design and may be constructed more rapidly and of cheaper materials, the work being done in small plants with simple facilities and using ordinary skilled labor. Also, they are easier to erect or to take down and are more readily repaired after a serious crash.

Airship Gases

Hydrogen and helium are the only known gases that are sufficiently light to be of practical value for use in airships. Under the standard atmospheric conditions commonly used in aerodynamic calculations (dry air at a temperature of 15 deg. cent. and a pressure of 760 mm. of mercury), the lifting power of pure hydrogen is 71.2 lb. per 1000 cu. ft. and that of pure helium 66 lb. In computing the gross lift of an airship it is customary to use somewhat lower figures to represent the "standard lift" of these gases. The values quite often used are 68 lb. per 1000 cu. ft. for hydrogen and 63 lb. for helium; there is some diversity of opinion, however, as to these figures. Although the lifting power of helium is about 7.3 per cent less than that of hydrogen, nevertheless helium is the ideal airship gas. It is absolutely inert, has no effect upon the materials with which it comes in contact, and is non-combustible. While hydrogen is considered one of the relatively inert gases in that it does not attack metals, it does have a very deteriorating effect on rubberized fabrics. It is quite inflammable and becomes dangerously explosive when it contains more than 15 per cent of air and hence must be handled with great care. Hydrogen can be produced at a cost of from $2.75 to $10 per 1000 cu. ft., depending upon the quantity, while the present (June, 1925) cost of helium, not including overhead charges, is from $55 to $60 per 1000 cu. ft. It is expected, however, with increased production and improved methods, to reduce the cost of helium to from $25 to $30 per 1000 cu. ft. Because of the deteriorating effect of hydrogen on the rubberized fabrics used in non-rigid and semi-rigid airships and the added wear and tear due to the increased number of deflations and re-inflations necessary when hydrogen is used, the Army Air Service has found it more economical to operate these ships with helium, even at its present price.

Thus far the only known source of supply of helium in commercial quantities is from the natural gases of the North American continent. The richest helium-bearing gas fields are those of the Mid-Continent, Texas, and Salt Lake regions. The average helium content of the gases from these fields is about 1.33 per cent. It is estimated that 500,000,000 cu. ft. of helium are going to waste annually in the United States. (See Eighth Annual Report of the National Advisory Committee for Aeronautics.) The helium is extracted from the natural gas by means of a continuous counter-flow liquefaction process in which the gas from the mains is passed through a refrigerating apparatus where all the component gases excepting helium are liquefied, the helium passing on to compressors which deliver it to special containers under high pressure. The helium-free gas is returned to the mains through heat exchangers where it absorbs heat from the incoming gas which has previously been compressed, as one of the initial steps of the first stage of the liquefaction process, to a pressure of 2000 lb. per sq. in. The only helium-extraction plant in the world—the United States Government plant at Fort Worth, Tex.—has a capacity of approximately 1,000,000 cu. ft. of helium per month and operates on gas from the Petrolia field. The gas from this field contains about 0.945 per cent of helium.

In order to minimize the likelihood of having to release gas, especially helium, on account of the ship becoming excessively light due to consumption of fuel, it is necessary to keep the ship in static equilibrium during flight. This may be accomplished by the condensation of a sufficient amount of the water of combustion from the engine exhaust gases to compensate for the weight of fuel consumed. For every 100 lb. of gasoline burned there are approximately 145 lb. of water vapor in the exhaust. The idea of recovering weight by condensing a portion of this water was conceived in England and tried out in the year 1909. The apparatus now used consists of a separate condenser, separator, and filter for each power

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plant. The weight of this additional equipment is more than compensated for by the decrease in the amount of ballast necessary to have on board at the beginning of a flight (ordinarily from 2.5 to 5 per cent of the gross lift for ships without ballast recovery). Service tests on the water-recovery apparatus of the U. S. Airship *Shenandoah* gave a recovery of 110 lb. per 100 lb. of gasoline consumed. In the case of airships inflated with hydrogen, static equilibrium may be maintained during flight by burning hydrogen in the engines with the liquid fuel in such proportions as to decrease the total lifting power by an amount equal to the weight of fuel consumed. The gas thus burned is replaced by fresh gas at the end of the flight. This system not only obviates the wasting of gas, but also keeps up the purity of the gas.

FORM, RESISTANCE, AND SPEED OF AIRSHIPS

Proper streamlining makes it possible to reduce the air resistance of an airship form, and consequently the power required to drive the ship through the air, to a remarkable degree, the resistance of the best airship forms being as low as 5 per cent of that of flat plates of like projected area. As the contained volume is the quantity which is fixed, since the total weight to be carried is known, the object of the designer is to select a form that will have minimum resistance for a given volume. Based on the results of tests made on airship models in aerodynamic laboratories, the fineness ratio, or ratio of total length to maximum diameter, should lie between 4.5 and 5.0, and the maximum diameter should be about one-third of the length of the ship aft of the nose. The U. S. Navy "C" class is one of the best of the American forms tested thus far. It has a fineness ratio of 4.62 and its maximum diameter is 30 per cent of the total length back from the nose. These tests also indicate that a length of parallel middle body up to one diameter may be inserted at the maximum diameter of the ship without any appreciable change in the resistance per unit volume. (N.A.C.A. Report No. 138.)

Early rigid airships had much larger fineness ratios than they should have had for good results and were parallel-sided for the greater part of their length, little attention being paid to streamlining other than the rounding or pointing of the ends. The first successful rigid, constructed by Count Zeppelin in the year 1900, had a fineness ratio of 11.05. As airship construction has advanced there has been a steady decrease in fineness ratios and great improvements have been made in hull forms. In rigids of the latest design the long parallel lines have given way to an almost perfect streamline form, the sharpness of curvature decreasing steadily from the bow to the stern. Such a form is used in the *Los Angeles*, which has a fineness ratio of 7.25, while the *Shenandoah* has a number of sections of parallel middle body and a fineness ratio of 8.64. When flying at full speed of 75 miles per hour the *Los Angeles* has a total air resistance of approximately 6900 lb., which gives an average thrust per engine of 1380 lb., with a corresponding propeller efficiency of 69 per cent. It is interesting to note that the two 5,000,000-cu. ft. commercial airships now under construction by the British Government are to have a fineness ratio of 5.26, less than one-half that of the first Zeppelin. This decrease in fineness ratios not only makes for increased aerodynamic efficiency but also for greater structural strength and hence for greater airworthiness; a short, fat ship is less likely to be subjected to damaging stresses than is a long, slender ship.

The relation between the horsepower, velocity, and total volume (air displacement) of an airship of streamline form may be expressed as follows:

$$\text{Hp.} = \frac{V^3 d(\text{Vol.})^{2/3}}{C} \quad [1]$$

where V = velocity in feet per second; d = density of atmosphere in slugs per cubic foot (1 slug = 32.17 lb.); Vol. = air displacement in cubic feet; and C is a constant ranging in value from 30,000 to 35,000 in a modern rigid airship and from 20,000 to 25,000 in small non-rigids of less than 200,000 cu. ft., volume. (N.A.C.A. Technical Note No. 194, by C. P. Burgess.)

In the design of large rigid airships it is assumed that V = 118 ft. per sec. = 80.5 m.p.h. = 70 knots; d = 0.00206 slug; C = 32,000. Whence, by substituting in Equation [1],

$$\text{Hp.} = 0.104\,(\text{Vol.})^{2/3} \quad [2]$$

(See Air Service Balloon and Airship School Instruction Pamphlet No. 8—Airship Operation, by Lt. H. H. Holland.)

From Equation [1] it is seen that the horsepower required to drive the ship varies directly as the cube of the velocity. Thus, in the case of the airship *Los Angeles*, with a full speed of 75 miles per hour with all five motors operating at rated load of 400 hp. each, if three of the five motors were put out of commission, leaving the ship with but 800 hp., she could still make a speed of approximately 55 miles per hour (73.5 per cent of full speed). Also, with one motor standing by in reserve and the other four operating at three-quarters rated load, giving a total of 1200 hp., she will cruise at approximately 63 miles per hour (84 per cent full speed).

The most profitable speed at which to operate an airship in commercial service on regular schedule depends upon a number of variable conditions. Herr W. Bleistein, in his paper on Effect of Speed on Economy of Airship Traffic (reprinted in N.A.C.A. Technical Memo. No. 302), gives the average value for this speed for airships having a volume of from 2,650,000 to 8,825,000 cu. ft.—the range of sizes of commercial interest today—as slightly over 60 miles per hour.

Again referring to Equation [1], for a given speed the horsepower required varies as the two-thirds power of the total volume of the ship. Since the maximum gross static lifting power varies directly as the volume of the ship, the horsepower required for a given speed varies as the two-thirds power of the total lift. Hence, with increase in size there is a decrease in the ratio of the total horsepower to the total lift of ships that are to operate at the same speed.

AIRSHIP ENGINES

As the airship is inherently a craft for operating on very long non-stop flights, its engines must necessarily be of the heavy-duty type capable of running several hundred hours without overhaul. The engines used on airships are, as a general rule, fitted with special pistons to give a somewhat lower compression ratio than when used in airplanes, and are operated at lower r.p.m. In laying out the power plant for an airship the number of separate units is made as small as is consistent with thorough reliability and proper distribution of weight, in order to keep the head resistance of the ship as low as possible and to reduce to a minimum the number of enginemen required on board. On very small single-engine airships the engine is carried in the after end of the control car; on small airships with more than one engine they are mounted on outriggers extending from the sides of the car (see Fig. 13) and so arranged as to make the engines quite accessible during flight; on the large ships the engines are carried in separate power gondolas, streamlined engine rooms (see Fig. 6), and are completely accessible, permitting of quite extensive repairs or replacements during flight. Furthermore, each engine operates with a skilled engineman in attendance at all times, thereby insuring the maximum degree of continuity of service. Large airships generally cruise with one or more engines standing by in reserve and with the remaining engines operating at about three-quarters rated load.

Great strides have been made in the development of aircraft engines since the war. Dependable, smooth-running, durable engines of rugged construction are now available in sizes up to 800 hp. in the water-cooled type and up to 400 hp. in the air-cooled type. There is a growing tendency in all aeronatuical work toward the use of air-cooled engines in sizes up to 300 hp. This means a very material saving in weight in view of the fact that the cooling system represents, on the average, something over 25 per cent of the weight of the water-cooled-engine installation. In the rigid airship, however, with its large enclosed heavy-duty power units, water-cooled engines are used exclusively. These engines, particularly in the larger sizes, are generally of the geared type to permit using propellers of greater diameter operating at lower speeds, thus making for greater efficiency. The propeller efficiencies attained in airship operation average from 65 to 70 per cent.

The airship propeller shown in Fig. 16 is constructed with a balsa wood core overlaid with mahogany plywood. It is 17 ft. 6 in. in diameter and weighs 140 lb.; it has a geometrical pitch of 13 ft. and a pitch ratio of 0.745; and is designed for use with a 600-hp. engine geared to give it a speed of 640 r.p.m. At a flying

FIG. 11 PROGRESS PICTURE TAKEN IN THE KEEL OF THE "SHENANDOAH," SHOWING TYPICAL STRUCTURE DETAILS AND A BATTERY OF FUEL TANKS (Note that provision is made for releasing tanks in case of emergency.)

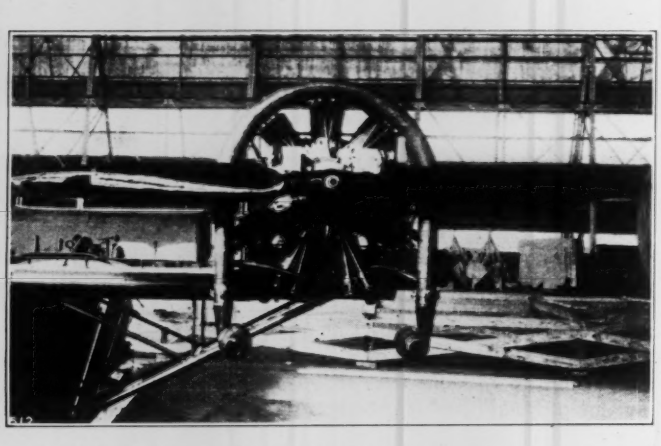

FIG. 13 WRIGHT MODEL J-4 200-HP. AIR-COOLED ENGINE MOUNTED ON THE PORT OUTRIGGER OF THE NON-RIGID ARMY AIRSHIP TC-6 AND FITTED WITH AN ALL-METAL PROPELLER

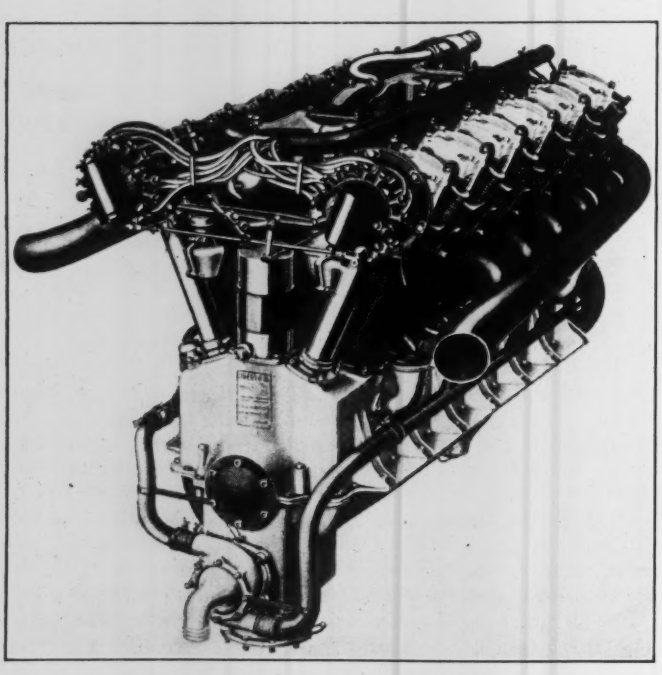

FIG. 14 THREE-QUARTER REAR VIEW OF THE 400-HP. LIBERTY-12 AIRCRAFT ENGINE

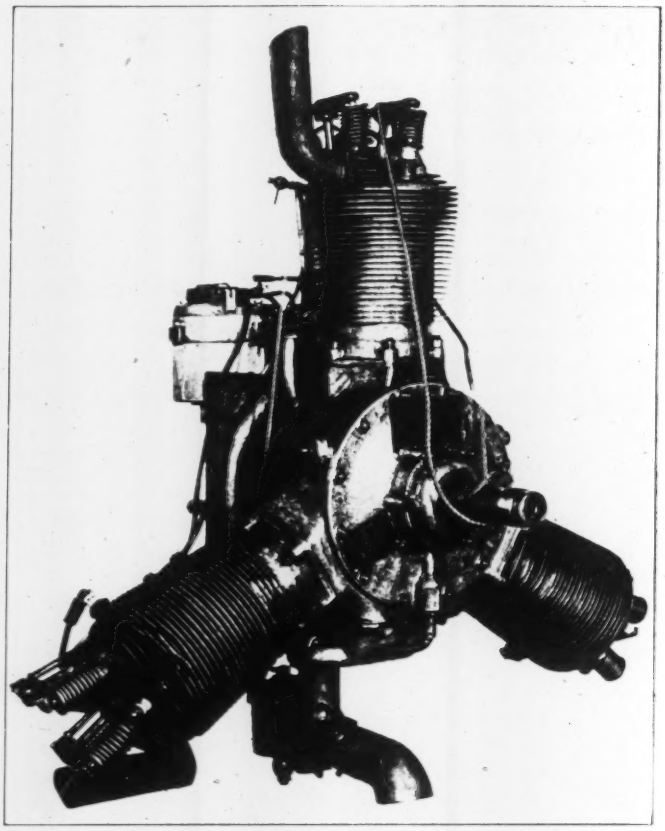

FIG. 12 WRIGHT MODEL L-4 60-HP. FIXED RADIAL AIR-COOLED ENGINE USED ON SMALL NON-RIGID TRAINING SHIPS

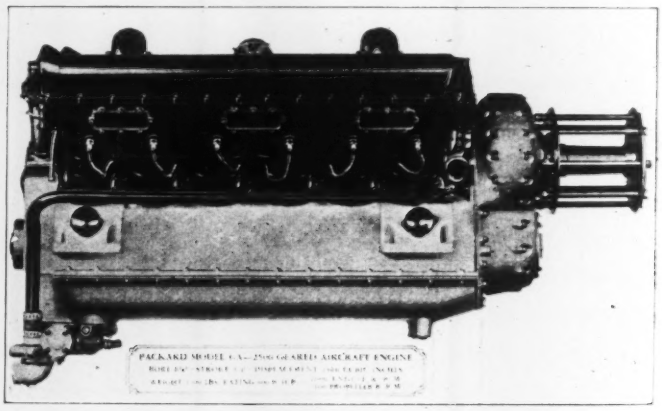

FIG. 15 PACKARD MODEL 1-A-2500 800-HP. GEARED AIRCRAFT ENGINE

FIG. 16 TYPICAL AIRSHIP PROPELLER FOR USE WITH A 600-HP. GEARED ENGINE, SHOWN IN COMPARISON WITH PROPELLER OF THE CURTISS PURSUIT PLANE DRIVEN BY A 400-HP. CURTISS D-12 ENGINE

speed of 70 miles per hour this propeller gives an efficiency of 70 per cent.

It is interesting to note that the 400-hp. Maybach special airship engines, with which the *Los Angeles* is equipped, have ball and roller bearings throughout; also that these engines are reversible, whereas the reversing of the propellers on most airship engines is accomplished through the medium of reversing gears in the transmission. It is understood that one of these engines was given a 300-hour, full-throttle, continuous-run block test at the factory in Germany and at the end of the test was found to be in almost perfect condition. These engines operate at a rated speed of 1400 r.p.m. with direct drive.

In order to eliminate the fire hazard accompanying the use of gasoline as fuel, especially that due to gasoline vapors which are bound to be present, particularly in warm weather, considerable work has been done in Europe on the development of heavy-oil (kerosene and heavier) engines and has met with good success.

Experience with mooring masts has demonstrated that the bow construction of the older ships has not been sufficiently strong to withstand the stresses to which the ships are subjected when riding at the mast in strong, gusty winds. The breaking away of the *Shenandoah* from the mast at Lakehurst, and of the British R-33 (of similiar bow construction) taught several valuable lessons regarding the bow construction of rigid airships and demonstrated the general airworthiness of such ships. Both ships remained aloft after the complete deflation of their forward gas cells (two cells in the case of the *Shenandoah*), rode out the storms, and came back to their hangars under their own power. The lessons learned from the experience with the *Shenandoah* were taken advantage of in designing the bow construction of the *Los Angeles*, which it is believed will outride on the mast any storm it is likely to encounter.

The Lakehurst mooring mast, which can be operated with from 9 to 12 men, is a steel structure 173 ft. in height. It is equipped with an electric passenger and express elevator, electric winches for handling the mooring cable and yaw lines, powerful flood lights, and with fuel, oil, water, and gas mains for serving the ships.

With the development of the mooring mast and the advancement that has been made in airship design and construction, the commercial airship will be placed in the hangar only when necessary for extensive repairs and overhaul. In other words, the hangar will bear about the same relation to the airship that the drydock bears to the steamship. This will result in a marked economic improvement in methods of landing—in the outlay of equipment and the personnel required. Mooring masts will be provided at all ports of call of large airships, including terminal stations, while hangars will be provided at terminal stations only.

TABLE 2 CHARACTERISTICS OF TYPICAL AMERICAN AERONAUTICAL ENGINES OF THE WATER-COOLED TYPE

Manufacturer	Model No.	Rated hp.	Rated r.p.m.	Wt. of dry engine, lb.	Wt. per rated hp., lb.	No. of cylinders	Hp. per cylinder	B.m.e.p. (rated load)	Bore, in.	Stroke, in.	Displacement, cu. in.	Angle bet. cyl. banks, deg.	Compression ratio	Oil con. per hp-hr. (rated load), lb.	Fuel con. per hp-hr. (rated load), lb.
Curtiss Aeroplane & Motor Co.	OX-5	90	1400	390	4.33	8	11.25	111	4.0	5.0	502.8	90	4.92:1	0.03	0.60
Wright Aeronautical Corp.	I (Low compression)	150	1450	467	3.11	8	18.75	117.3	4.7	5.1	718.88	90	4.72:1	0.023	0.515
Wright Aeronautical Corp.	E-4 (High compression)	190	1800	467	2.46	8	23.75	122	4.7	5.1	718.88	90	6.0:1	0.025	0.50
Curtiss Aeroplane & Motor Co.	D-12 (Low compression)	375	2000	693	1.85	12	31.3	133.8	4.5	6.0	1145	60	5.3:1	0.015	0.54
U. S. A.	Liberty-12 (Low compression)	382	1600	843.69	2.21	12	31.83	114	5.0	7.0	1648.9	45	4.9:1	0.03	0.50
U. S. A.	Liberty-12 (High compression)	400	1700	843.69	2.11	12	33.33	119.5	5.0	7.0	1648.9	45	5.42:1	0.032	0.496
Curtiss Aeroplane & Motor Co.	D-12 (High compression)	400	2000	694	1.73	12	33.33	137.5	4.5	6.0	1145	60	6.0:1	0.01	0.51
Packard Motor Car Co.	1-A-1500	510	2100	738	1.45	12	42.5	129	5.375	5.5	1499.45	60	5.5:1	0.015	0.50
Wright Aeronautical Corp.	T-3 (Low compression)	575	1800	1160	2.02	12	47.92	131.5	5.75	6.25	1947	60	5.5:1	0.025	0.50
Wright Aeronautical Corp.	T-3 (High compression)	675	2200	1160	1.72	12	56.25	136	5.75	6.25	1947	60	6.5:1	0.025	0.49
Packard Motor Car Co.	1-A-2500	800	2000	1100	1.38	12	66.67	132	6.375	6.5	2500	60	5.5:1	0.045	0.50

NOTE.—Compiled from data supplied by Engineering Division, U. S. Army Air Service.

TABLE 3 CHARACTERISTICS OF TYPICAL AMERICAN AERONAUTICAL ENGINES OF THE FIXED RADIAL AIR-COOLED TYPE

Manufacturer	Model No.	Rated hp.	Rated r.p.m.	Wt. without oil, lb.	Wt. per rated hp., lb.	No. of cylinders	Hp. per cylinder	B.m.e.p. (rated load)	Bore, in.	Stroke, in.	Displacement, cu. in.	Angle bet. cylinders, deg.	Compression ratio	Oil con. per hp-hr. (rated load) lb.	Fuel con. per hp-hr. (rated load), lb.
Wright Aeronautical Corp.	L-4	60	1800	160	2.67	3	20	120	4.25	5.25	223	120	5:1	0.03	0.50
Wright Aeronautical Corp.	J-4	200	1800	480	2.40	9	22.2	116.5	4.5	5.5	787.25	40	5:1	0.026	0.52

NOTE.—Compiled from data supplied by Engineering Division, U. S. Army Air Service.

The Bureau of Aeronautics of the United States Navy is also working along this line.

HANGARS AND MOORING MASTS

The operation of maneuvering a large airship into or out of a hangar in a strong, gusty wind is a difficult operation and requires a large number of men and a great deal of care and skill, particularly if the wind velocity exceeds 20 miles per hour. For wind velocities exceeding 35 or 40 miles per hour, even though blowing parallel to the hangar, the maneuver should not be attempted. The most difficult conditions to contend with are those accompanying a cross-hangar wind, especially if the angle between the wind direction and the axis of the hangar exceeds 45 deg. Under these conditions it would hardly be practical to attempt housing a large airship if the wind velocity were greater than 8 or 10 miles per hour. Rather than attempt to put an airship into a hangar under adverse wind conditions, it is better to take to the air and await a more favorable time or else moor the ship to the mooring mast.

The mooring mast, which was first developed to a point of successful operation by the British, solves one of the most serious problems the airship has had to contend with—that of mooring in windy weather. Quoting Sir Sefton Brancker, former British Air Minister, "The fact that airships now need not be housed in sheds except for the purpose of serious overhaul is perhaps the greatest step in progress accomplished in the whole history of airships."

EFFECT OF WEATHER ON THE OPERATION OF AIRSHIPS

The difficulties experienced in getting large airships in and out of their hangars in windy weather in the days before the advent of the mooring mast when it was necessary to house the ships after each flight, led to the idea that the airship was strictly a fair-weather craft. This, however, is not a true conception. With a reasonable reserve of power and fuel the airship need not fear any kind of weather.

Hail, rain, snow, and fog cause no serious trouble. With modern navigating instruments and methods it is possible to maintain an accurate course without seeing the ground. The velocity of the ship through the air keeps it quite free from any accumulation of snow or hail, and the protective coatings applied to the fabric of the outer cover reduce the absorption of rain to a minimum. Such increase in weight as may be occasioned by the rain is taken care of by the dynamic lift. In the case of a large ship the added

weight due to rain on the hull may amount to several tons. In approaching an airport in dense fog, clouds, or a snow storm, the airship can advance as slowly as necessary and can hover over the field, communicating directly with the landing party to make sure that all is in readiness and that the way is clear of obstructions.

Wind (horizontal movement of air relative to the earth) results only in increasing or decreasing the ground speed of the ship, and is dangerous only in that it may delay the ship beyond the limits of its fuel supply. It is customary to allow a fuel reserve of from 25 to 50 per cent, depending upon conditions, to cover this contingency.

The danger from lightning when flying in electrical storms is not serious in the case of helium-filled ships; in the case of a ship inflated with hydrogen the pilot must see that no gas is released while in the vicinity of such storms, as there seems to be evidence that a rising column of hydrogen is a conductor of electricity and that there is danger of an atmospheric discharge traveling down this conductor to the interior of the gas trunks and causing a serious explosion. It has been suggested that, as a precautionary measure against such an occurrence, a continuous flow of cooled exhaust gases from the engines be maintained up through the trunks in order that any hydrogen released into the trunks may at all times be diluted with inert gas instead of air. In fabricating rigid airships, regardless of whether hydrogen or helium is to be used, great care is exercised to obtain a perfect electrical bond between all structural members in order that the metal framework may form an unbroken conductor, thereby permitting a charge of lightning to pass through the structure without interior sparking. A number of instances are on record of Zeppelin airships inflated with hydrogen being struck by lightning while in flight without suffering any damage; fused spots were found at the points where the current entered and left the ships.

The vertical air currents and eddies accompanying electrical storms are the most serious conditions with which the airship has to contend. These currents are strongest immediately in front of and behind line squalls and thunderstorms, and at times reach high velocities. The danger under these circumstances, unless the ship is skilfully handled, is that her structure may be subjected to extreme stresses, that she may be carried to such a height as to lose an excessive amount of gas through her automatic valves and that she may be forced down against the ground. Such air conditions are met by keeping the ship trimmed level and in a state of static equilibrium, thereby insuring maximum response to the controls, the action of the vertical currents being counteracted by the proper use of the control surfaces and the motors.

The area, intensity, and movement of storms are shown on the daily weather maps and broadcast in the radio weather reports and storm warnings; thus, in case of an approaching storm, the pilot may alter his course in advance in such a manner as to avoid that portion of the storm which unduly would delay his progress and, in many cases, to pass on such side of the storm as to take advantage of favorable wind conditions.

NAVIGATION OF THE AIRSHIP

The methods used in aerial navigation are the same in principle as those employed for years in marine navigation, although the practical application of these methods has necessarily been modified to suit the requirements of aircraft. With the modification of methods and instruments and the development of special instruments—such as the drift and ground-speed indicator and earth inductor compass—it has become possible to navigate aircraft with a surprising degree of accuracy. Nevertheless, much remains to be done to bring the science of aerial navigation to the high state of perfection it is hoped eventually to attain.

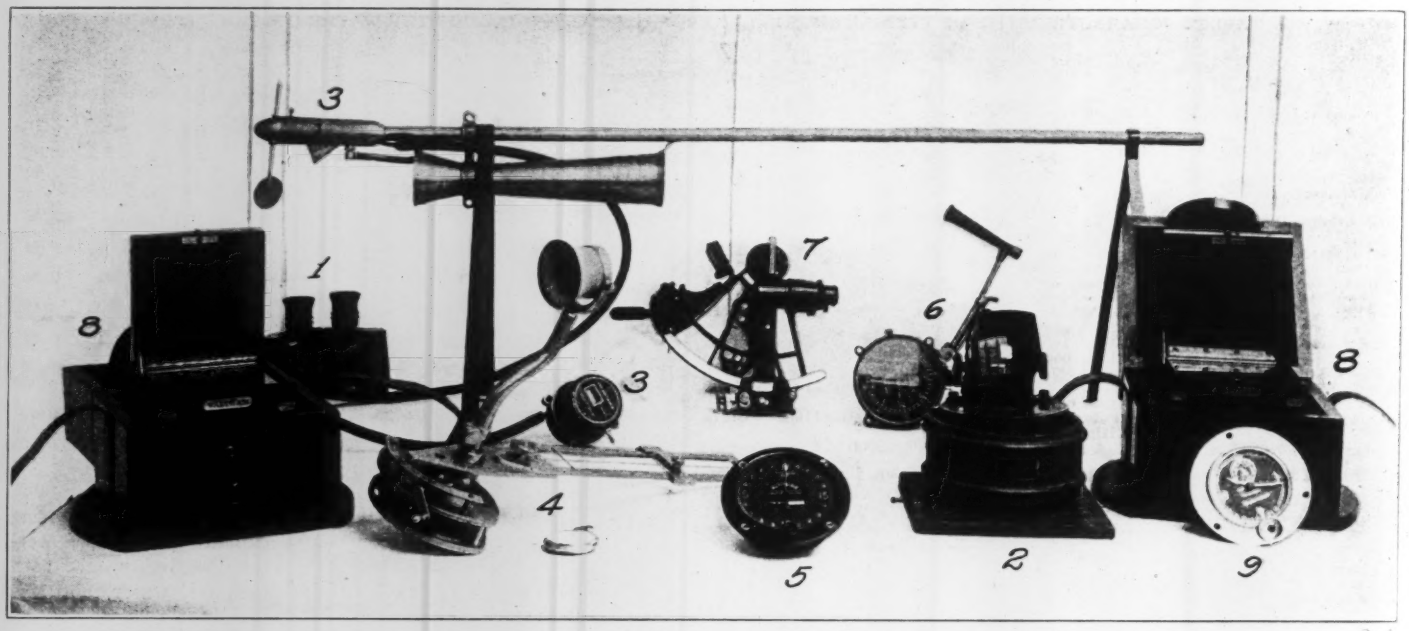

FIG. 17 GROUP OF TYPICAL AERIAL NAVIGATING INSTRUMENTS FOR AIRSHIP USE

(1, Binoculars; 2, Aircraft magnetic compass; 3-3, Air-distance recorder; 4, Drift and ground-speed indicator with stop watch; 5, Altimeter; 6, Air-speed indicator with its pitot-venturi nozzles; 7, Aircraft bubble sextant; 8, Chronometers, one set to Greenwich mean solar and the other to Greenwich sidereal time; 9, Aircraft clock.)

There are four fundamental methods employed in aerial navigation—pilotage, dead reckoning, astronomical observation, and radio bearings—no one of which is applicable at all times or sufficient in itself. The method or methods to be used in any particular case must be determined by existing conditions. In general, the pilot proceeds by pilotage or dead reckoning, as circumstances may dictate, plotting a continuous record of the ship's position on his chart and checking and correcting the position at frequent intervals by astronomical observations or radio bearings.

THE AIRSHIP IN COMMERCE

The airship and the airplane possess characteristics which give to each certain advantages over the other, characteristics which cut out for each a very definite service in the sphere of commercial aeronautics.

The airship has a much greater non-stop cruising radius than the airplane; the most efficient distance between ports of call for the commercial airship, allowing a 50 per cent fuel reserve, being from about 2000 to 4000 miles, depending upon the size of the ship and upon traffic and operating considerations. In the case of the commercial airplane this distance ranges from about 300 miles to possibly 600 miles. Between points of call on the airplane route, or airway, must be located suitably marked emergency landing fields for use in case of forced landings. On the United States Air Mail routes, particularly where night flying is to be done, these emergency landing fields are located at approximately 25-

mile intervals through the flatter, more open country and at 10-mile intervals through hilly country where natural landing fields are difficult to find.

The airship can carry much heavier loads than the airplane. It can reduce its air speed to zero and still remain in the air; and can travel and land in fog, snow storm, and night with a greater degree of security than the airplane. Furthermore, with this ability to remain in the air when not under power and with its relatively large number of heavy-duty power plants, all of which are completely accessible during flight and operate under the continuous attention of skilled motor mechanics, the airship offers a greater degree of safety and reliability. Another point in favor of the airship is that it affords greater freedom of movement and comfort to its passengers, offering far more commodious and convenient accommodations, free from noise and vibration, for travel either by day or by night. The provision of suitable sleeping accommodations for passengers on airplanes, of the present type at least, is quite a serious problem and materially reduces the number of passengers that can be carried.

airship and the airplane, the future development of commercial aeronautics will be divided into two distinct branches, as follows:

(1) Large rigid airships carrying heavy loads of passengers, mail, and express on very long non-stop flights, particularly over transoceanic or combination transoceanic and transcontinental routes, where the saving in time over surface transportation will be a maximum, with terminals and ports of call at the great traffic centers of the world.

(2) Airplanes carrying lighter loads of passengers, mail, and express at higher speeds over routes which can be provided with a system of intermediate landing fields interspersed with a chain of emergency fields. Some of these airplane lines will operate as feeders and distributers for the airship lines, while many will operate entirely independently.

The smaller type of airship, either non-rigid or semi-rigid, is well suited for long-distance reconnaissance, photographic, and forest-patrol missions over rough mountainous country where no landing fields are available and where it may be desired to fly

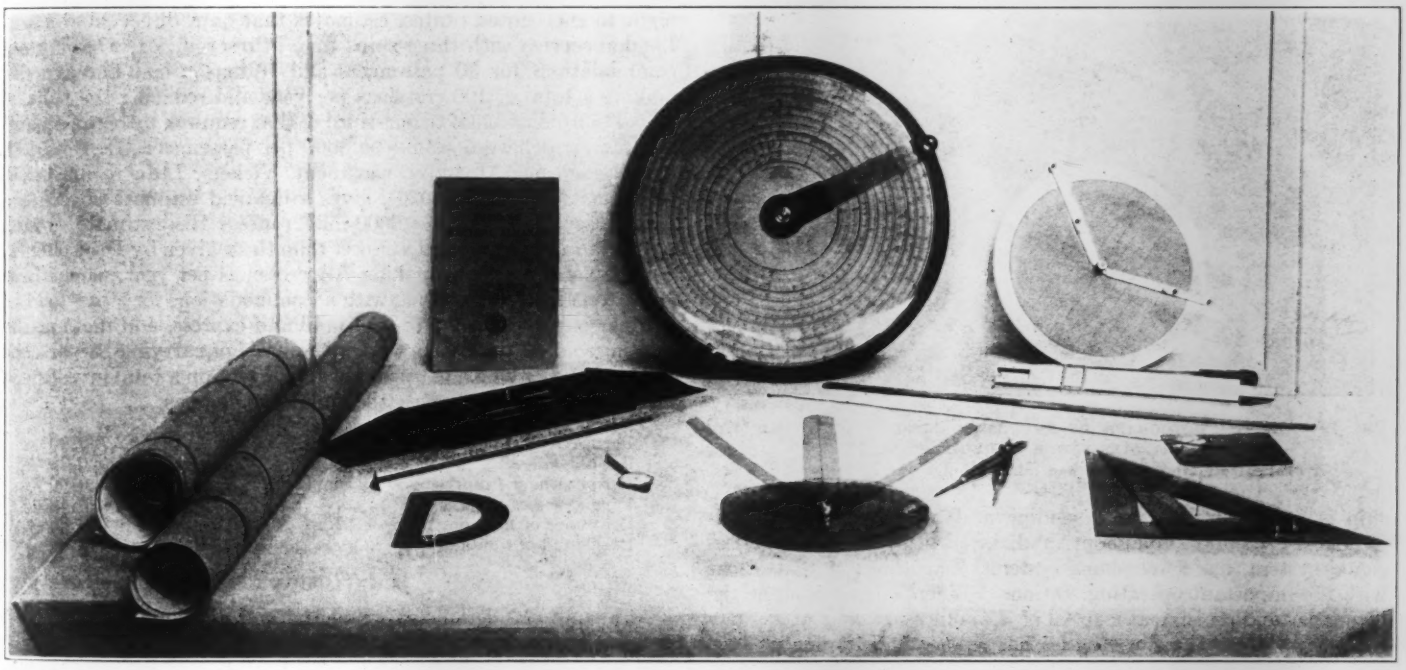

FIG. 18 MAPS, CHARTS, NAUTICAL ALMANAC, COMPUTING AND PLOTTING INSTRUMENTS, ETC., USED BY NAVIGATING OFFICER

The airplane, on the other hand, travels with much greater speed than the airship and is very easily housed. Being of smaller size it is cheaper to construct and to maintain, which makes possible the establishment of commercial airplane lines, on a moderate scale at least, with a much smaller initial investment than is possible with airships.

For airships of similiar form the horsepower required, and hence the total weight of power plants, for a given speed varies as the two-thirds power of the maximum gross lift, since the maximum gross lift varies directly as the total volume. Hence, with increase in size there is a decrease in the ratio of the total weight of power plants to the total lift of the ship. For a given speed and cruising radius the pay load increases quite rapidly with increase in size up to a certain point, given by G. A. Crocco (see N.A.C.A. Technical Memo. No. 274) as about 1000 metric tons total lift, beyond which point there is very little increase. In the case of the airplane the lift-drag ratio has a tendency to decrease with increase in the size of the plane, which means that the drag increases at a somewhat more rapid rate than the lift. Hence, for a given speed the horsepower required tends to increase more rapidly than the lift with increase in size of the plane. The ratio of the useful load to the gross lift of the airplane increases with increase in size up to a certain limit and then begins to decrease. Prof. E. P. Warner, in his article on The Limiting Range of Airplanes, places this limit at a gross weight of about 10,000 lb.

On account of the differences between the characteristics of the

very slowly at times, or even hover over certain sections, in order to make a careful study of the terrain, as in railroad and transmission-line location, water-power developments, etc.

The Army Air Service has been doing experimental work both with the idea of using airships as airplane carriers and of using airplanes for carrying mail, express, and even passengers to and from the large commercial airships in flight as they speed over important cities on their long non-stop flights. The maneuver of picking up an airplane in flight in mid-air, carrying it, and launching it again was accomplished for the first time by the Army Airship TC-3 on Dec. 15, 1924, at Scott Field. The test was made at an altitude of about 1500 ft. and at an air speed of 60 miles per hour. The plane, a Sperry Messenger, flew up under the ship, made contact and was carried 10 or 15 miles before launching away. The test was entirely successful and is undoubtedly a forerunner of some very practical developments.

The British Government now has under construction two commercial rigid airships, each having a gas displacement of 5,000,000 cu. ft.—approximately twice that of the *Los Angeles*. According to data published in current European and American aeronautical journals, each of these ships is to have a length of 695 ft., a maximum diameter of 132 ft., seven 550-hp. engines, a maximum speed of 80 miles per hour, a cruising speed of about 65 miles per hour, and accommodations for a pay load of 120 passengers as well as 10 to 15 tons of mail and express. In addition to commodious passenger cabins, there are to be provided a social hall, smoking room, excel-

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lent dining facilities, and quarters for a crew of from 35 to 40 men, including cooks and stewards. Since hydrogen is to be used as the lifting gas, the Ricardo kerosene-cum-hydrogen fuel system is to be employed, the passenger quarters are to be separated from the rest of the ship both by gastight walls and by a free air space, and it is reported that cooled exhaust gases from the engines are to be circulated through the trunks from the gas valves. The contract price for these ships is given as £350,000 ($1,680,000) each. They are to operate between England and Australia, stopping en route in Egypt, India and Java—the total distance of some 12,000 miles to be covered in 11 days, whereas the time required for this journey by ocean liner is from 32 to 36 days.

The *Los Angeles*, the most modern airship in the air today, has accommodations for from 24 to 30 passengers and from 2 to 2.5 tons of mail and express, the total pay load being about 6.4 tons. The passenger compartments and conveniences are comparable in many ways to those to be found on a standard Pullman car. The

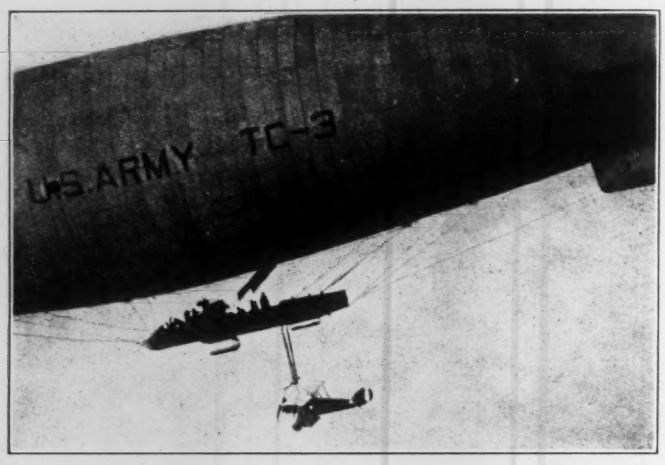

FIG. 19 AERIAL PHOTOGRAPH OF THE ARMY AIRSHIP TC-3 AND THE SPERRY MESSENGER ARMY AIRPLANE
(Taken just after the two had made contact during flight.)

ship is equipped with radio sending and receiving apparatus, is electrically lighted throughout, and, in addition to a mechanical signal system, has a telephone system connecting the control car with the important operating stations. When using helium as the lifting gas and cruising at a speed of 57.5 miles per hour with a pay load of 6.4 tons, the *Los Angeles* has a maximum radius of 3450 miles, which, on the basis of a 50 per cent fuel reserve, gives a commercial cruising radius of 2300 miles. A number of commercial routes have been suggested for this ship, one of which is the New York-London route, going via Newfoundland and Ireland and returning via the Azores and Bermuda, thereby taking advantage of prevailing winds; other suggested routes are the New York-Panama and the Los Angeles-Honolulu.

COST CONSIDERATIONS

The cost of airship construction depends upon a number of conditions, such as the size and number of ships to be built, the facilities available for carrying on the work, previous experience in airship construction, and the labor and materials market. On a gas-displacement basis this cost ranges from about $0.35 to $1.50 per cu. ft. While the actual cost of the *Los Angeles* to the German Government has not been made public, it is estimated that it was approximately $1,000,000, which on a gas-displacement basis represents a cost of $0.384 per cu. ft. The contract price on the new 5,000,000-cu. ft. British ships is $0.336 per cu. ft. According to testimony given before the Naval Appropriations Sub-Committee by Rear-Admiral Wm. A. Moffett, Chief of the Bureau of Aeronautics, the Navy could construct a single experimental rigid airship of 6,000,000 cu. ft. gas capacity at a cost of $1 per cubic foot, and by constructing four such ships the cost would be reduced to one-half that amount. The *Shenandoah*, the first rigid airship to be built by the Navy Department, cost about $1.37 per cu. ft.

Regarding the rate of depreciation of commercial airships operating on regular scheduled service, the data available are very limited. It is known, however, that rigid airships put into service on commercial lines in Germany in the year 1910 were still in operation in 1914 when the war broke out; hence, it would seem safe to assume a useful life of four or five years, if not longer.

Little information is to be had at the present time concerning the actual cost of operating airships on a commercial service. The German Air Transport Company, which operated four 30-passenger hydrogen-filled Zeppelin airships on regular service for several years before being stopped by the war, again inaugurated, in 1919, a commercial line operating the Zeppelin airship *Bodensee*, with accommodations for thirty passengers, on a regular passenger and mail service between Berlin and Friedrichshafen. This line was soon forced to discontinue, under the terms of the Versailles Treaty. So far as the author has been able to learn, no data regarding the operation of these ships have been made public other than that some 35,000 passengers and many tons of mail and express were carried without accident or loss of life; it does not seem likely that such service would have been resumed had it not proven profitable.

Dr. Hugo Eckener, who commanded the *Los Angeles* on her flight to the United States, estimates that on a direct New York-London service with three ships (one in reserve), each having accommodations for 30 passengers and 10 tons of mail and express, making a total of 100 crossings per year and reducing the time of travel to from one-half to one-third of that required by ocean steamers, the fare charged would be $600 per passenger. H. B. Pratt, Chief Engineer, Airship Department, Vickers, Ltd., in his book, Commercial Airships (1920), gives a detailed estimate covering a similar service over this 3500-mile route. His estimate, which places the charges somewhat lower than those given by Dr. Eckener, contemplates a service making 104 crossings per year, using three ships (one in reserve), each with accommodations for a pay load of 100 passengers and 10.5 tons of mail and express, and making the trip in from 50 to 60 hours. On the basis of carrying 83 per cent full load and paying a net profit of 10 per cent on a total investment of $12,000,000, he derives the following charges:

Per ton from London to New York	$2,760.00
Per ton-mile	0.79
Per passenger from London to New York	369.60
Per passenger-mile	0.105
Per ounce of mail matter	0.08
Per pound of mail matter	1.28

CONCLUSION

Although considerable experimental and development work still remains to be done to bring the large rigid airship to the highly developed state of the ocean liner, nevertheless, within the course of a few years this type of airship, especially when using helium as the lifting medium, will be classed as a safe, reliable, time-saving, comfortable craft, admirably adapted to carrying heavy loads of passengers, mail, and express on profitable commercial service over long non-stop flights. When augmented by the proper ground organization, including terminal facilities, mooring masts, radio stations and meteorological service, the rigid airship will be capable of maintaining regular scheduled service to any part of the globe.

The United States enjoys the unique position of having, as one of its natural resources, an abundant supply of helium. This gives us a great advantage over the other countries of the world in the operation of airships, both as commercial carriers and as instruments of national defense. At present we are wasting this valuable asset at an appalling rate, the estimated annual waste being 500,000,000 cu. ft. How long this supply will last we do not know. We do know that it cannot be expected to continue at this rate indefinitely, and that all reasonable steps should be taken to safeguard and conserve the supply.

The airship and the airplane are fitted by the characteristics peculiar to each for service in separate fields of the sphere of commercial aeronautics. Each has its own particular mission to perform and its own contribution to make toward the advancement of civilization by shortening distances and bringing the world closer together. The two types of aircraft must work together, each receiving the hearty coöperation and support of the other, if commercial aeronautics is to render to society its full measure of service.

Materials-Handling Problems and Their Solution

A Paper Dealing with Problems of Movement of Products in Industrial Plants, Whether Goods are Being Received or are Going through the Various Processes of Operation, Distribution, Assembly, Collection, Packing, Storage, and Shipment, and the Application of Mechanical Means to Effect Such Movement

By F. D. CAMPBELL,[1] NEW YORK, N. Y.

IT IS BELIEVED that the greatest general interest in movement problems lies in the handling of the great miscellany of packages, or articles unenclosed, in some definite form or unit, as distinguished from loose bulk material. Space is not available for dealing with the entire subject of materials handling, and as it has been the practice to handle bulk material mechanically for many years, it is, comparatively speaking, not so much of a materials-handling problem, in that its applications are quite well known.

Machine processes are today in a high state of development, and if materials handling were developed to the same high plane it would be little in the way of a problem in manufacture.

Granting that its processing machinery has been well selected and is most suitable for the manufacture of the product, and that personnel, organization, and sales are on the best possible plane of efficiency under the circumstances, with cost systems and offices functioning properly, if more profits are to be gotten out of a business, a fertile field for cultivation is to be found in organized handling methods.

Ordinary Examples of Handling Problems and Their Solution

The history of a certain moderate-sized plant over several years developed on analysis the fact that profits and expansion were subnormal. Expense kept pace with sales proportionately in dollars and cents. The entire structure of the business was well scrutinized and it was found to be in good shape throughout except in that portion involving the handling of the goods. Here was the "wolf at the door." There was no sequence to movement, no related locations of the various operations. Excessive space was used for goods in transit, resulting in low productive value, and there were many handling operations due to the arrangement, thus necessitating much labor for these many motions.

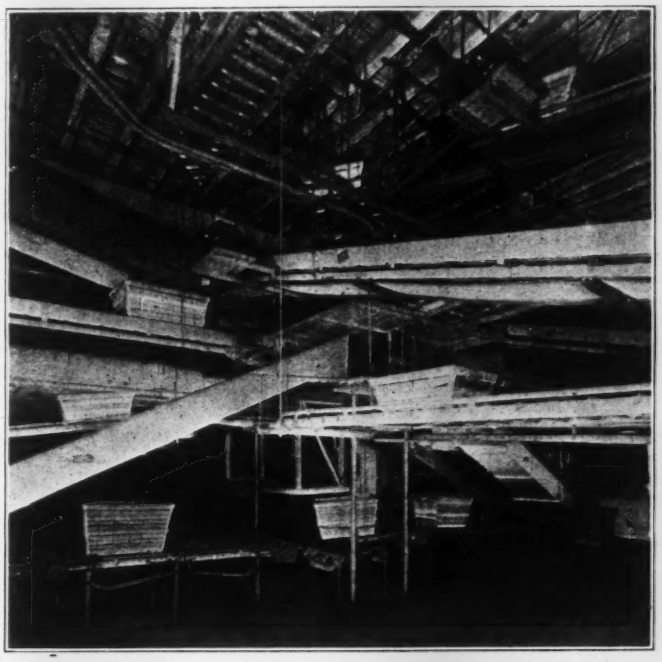

Fig. 1 An Example of Many Transit Lines, Collecting and Distributing Containers, and Occupying But Very Little Floor Space

(The goods of the entire establishment are sent through this point and selectively despatched to required areas by a single operator, and empty containers are returned to sources. Sixteen transit lines with a total of twenty-six terminal points are involved in this illustration.)

In consequence, the time element of travel was at least four times that absolutely required. The business was just about self-supporting, but not paying dividends. Added capital would not have been of assistance as there was no sound basis for its employment under existing conditions, but it would have been justified if the ratio of income to outgo could have been changed.

The result of the study was that a complete rearrangement was planned which included all necessary handling equipment, and a project was financed to put it into immediate operation. The first year's operation returned more than the initial outlay after deducting fixed charges. The second year's handling expense was practically no greater than that of the first year, though there was a healthy increase in the volume of business.

The change in the ratio of profits to expense was such that a new program for healthier expansion was possible, and thus the business was literally put on its feet by the use of proper methods of handling. It was planned to amortize the cost of the project, approximately $100,000, in ten years, but the actual returns from two years' operation indicated that it would earn a million dollars in this period, or a net return of the initial outlay annually.

Another instance where new handling methods proved to be a large source of profit was one where expansion in quarters was apparently necessary to handle increased business. Careful examination disclosed the fact that much of the floor space of the plant was unproductive, however crowded it appeared from casual observation. There was a distinct lack of fluidity of material movement through the plant. Much congestion resulted from large stocks of materials in varying stages of process being placed conveniently here and there to keep the processing equipment in work. Large aisles were necessary for the back-and-forth movement in many directions, and supervision was costly, as was also the burden for carrying stocks at a continuously high inventory.

A rearrangement was effected that in its very simplicity corrected the troubles. Line flow in sequence was provided, with only sufficient immediate stocks on hand for the minimum elasticity required and to keep pace normally with production orders. The reduction of local storages and aisleways thus brought about approximately a 50 per cent shrinkage in floor space. The "stock in process" inventory was reduced about 75 per cent, the unskilled labor used for handling was almost entirely done away with between processes, any selective change of pace was permitted for seasonal peak periods with capacity for over 100 per cent increase in output, the manufacturing period was reduced very materially, and an order and rhythm introduced that greatly reduced supervision costs.

The product in question is a mechanical device built in a number of types and sizes, and many of its component parts and basic products came from different places and from some distance. As noted above, this example furnishes evidence of almost every benefit possible from the introduction of new methods of arrangement and the incorporation of handling devices. An automatic conveyor system was the means to this end. The space vacated permitted the emplacement of other contributing departments and so increased the productiveness of the total floor space that several times the initial outlay was returned.

On examination and survey of the average run of manufacturing plants it will be found that a great number of conditions exist that directly or indirectly contribute to high manufacturing costs, and

[1] Industrial Engineer, Mem. A.S.M.E.

Contributed by the Materials Handling Division for presentation at the Annual Meeting, New York, November 30 to December 4, 1925, of The American Society of Mechanical Engineers. All papers are subject to revision.

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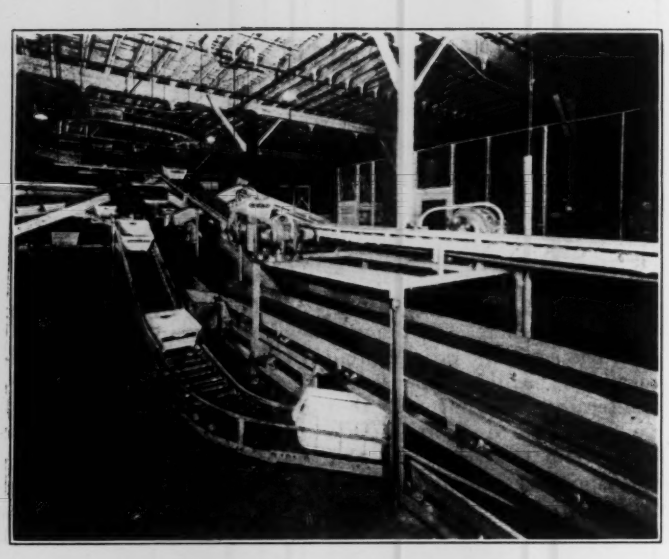

FIG. 2 AN ILLUSTRATION OF ECONOMY IN EQUIPMENT

(An automatic conveyor in the foreground carries two separate streams of goods on the upper side and two streams in the reverse direction on the lower, while supporting an additional line above it. Feed lines are shown returning empty containers.)

FIG. 3 AN ILLUSTRATION OF MECHANICAL HANDLING THROUGH PROCESS

(Continuous streams of crates and containers holding bottled goods are being automatically carried out from filling machines, with but three operators engaged in the filling and movement of the product. Processing units are permitted to produce at maximum speed without congestion and no labor is required for movement.)

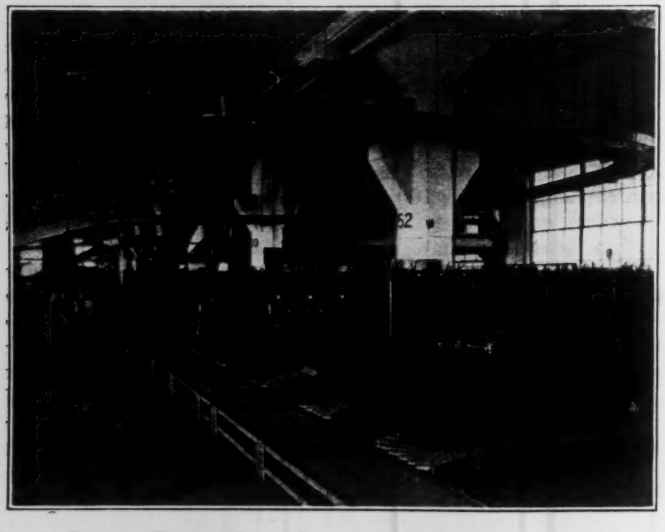

FIG. 4 PROCESS TRANSIT LINES SERVING TEST PANELS WITH COLLECTING CONVEYOR IN FOREGROUND, WHILE THROUGH LINES ARE CARRIED OVERHEAD FOR OTHER DISTRIBUTION

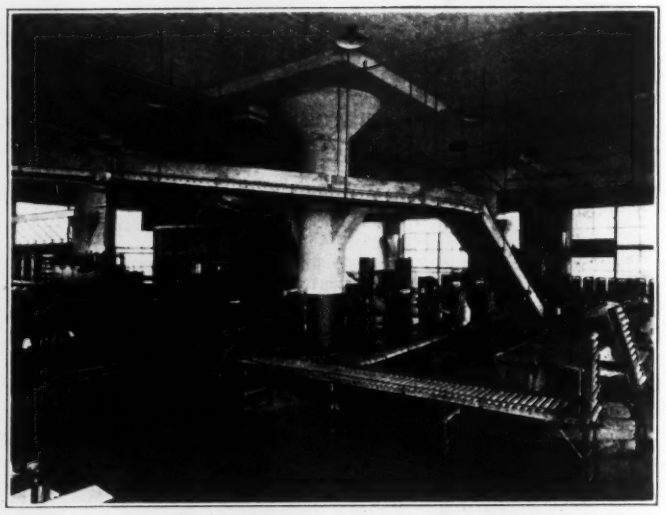

FIG. 5 THE ROLLER WORK BENCH WHERE ARTICLES OF CONSIDERABLE WEIGHT ARE EASILY MOVED FROM ONE OPERATION TO ANOTHER IN A STEADY STREAM AND IN UNIFORM TIME ELEMENTS

(The work must proceed in an orderly procession, with no opportunity for congestion or accumulation about operations. A steady pace is maintained while parts that are to be added are delivered to the area by overhead transit lines.)

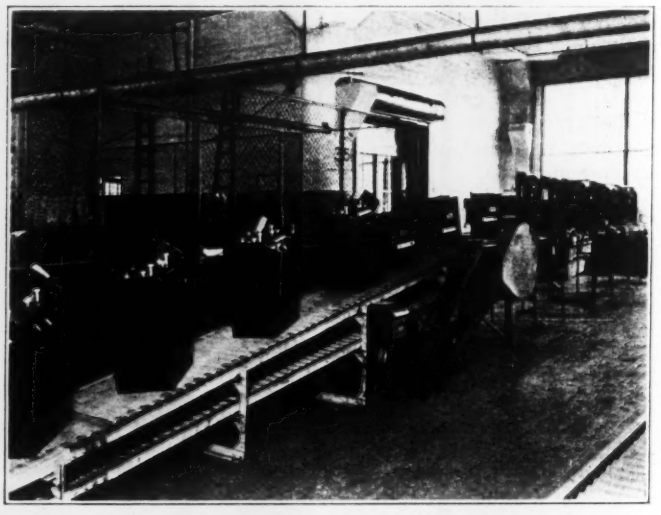

FIG. 6 A TRANSIT CONVEYOR LINE FOR HANDLING HEAVY ARTICLES FROM ONE DEPARTMENT TO ANOTHER. AN EXCELLENT EXAMPLE OF STREAM MOVEMENT OBTAINED AUTOMATICALLY

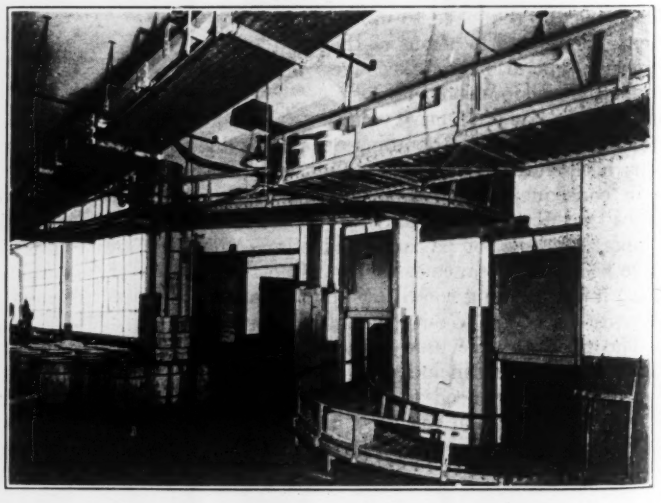

FIG. 7 AN ILLUSTRATION OF OVERHEAD TRANSPORTATION WITH SELECTIVE DIRECTION, WHILE FLOORS ARE CLEAR BELOW

(A profitable reclamation of floor area for productive purposes that was originally burdened with transit of goods.)

the examples cited above are typical, indicating where sources of unwarranted costs may be discovered.

PRESENT ARRANGEMENT OF MANY PLANTS AND THE RESULT

Many manufacturing plants are laid out and built without regard to handling problems. Buildings are designed to suit certain ground property and the pocketbook. Space is allocated for offices, manufacturing, stocks, and shipping, with the various departments or sections scattered here and there, on a Yankee guess as to requirements. The whole layout has little regard for sequential movement, nor are departments located in sequential relation to each other. Likewise storage of goods in any state is made in the odd corners of space thought to be of the least productive value. Distribution is through a labyrinth of routes and handling is a helter-skelter affair, criss-crossing and frantically endeavoring to keep up with the demand to get something in or take it out. Storage, local to process, becomes excessive to save delays, and units of movement become heavy trip loads. This necessitates much free way for aisles, but never overcomes the choke points and general congestion at peak periods. Some attempts are made to schedule the movement, but they do not usually result in any kind of balance. Not only is much unskilled labor necessary for this handling, but skilled workers are forced to perform unskilled work in moving materials to and from the machines they operate. Skilled labor thus adds this burden to the operation cost.

Ninety per cent of the industrial plants of the country are burdened with handling difficulties, though they may not be particularly evident to the managers of these plants. The search for places in which to effect cost reduction goes on in every direction save that of materials movement. If a possible saving in handling does suggest itself, the thought often leads them apparently to introduce so many changes that the whole matter winds up in a maze and is dropped as being impractical or too intricate for solution. But there is much haste to dam these heretofore undiscovered leaks when once the manufacturer finds he can obtain an authoritative estimate of the earnings that materials-handling equipment makes possible through the reduction of labor, supervision, inventory, space, and manufacturing period, and in addition get the benefits of better production control, flexibility to meet seasonal market demands, and quicker deliveries.

SUGGESTED SOLUTIONS OF THE HANDLING PROBLEM

In the solution of the handling problem the principle of a watershed system is adopted, but in a manner to suit the premises under consideration as well as individual requirements. Progress entirely by gravitation is not always possible, and power is sometimes necessary for boosting or for changing from one level to another. The flow, however, no matter how circuitous, may still be in line in effect.

The application of mechanics to the handling of products first has its conception on paper in a graphic method of procedure, adapting the flow to the premises, or in the case of a new plant, adapting the premises to the flow if possible.

The individual worker is a self-directed unit and proceeds anywhere at will, but carrier devices are installed like a railroad, and sources or process points are ranged along the carrier line, which may make as many turns as desired to take advantage of all productive space. In doing this some thought must be given to the necessity for ample capacity, varying production rate, development of new lines, changeable character of product, and expansion to take care of a growing business.

Arrangement is paramount with regard to the transporting devices used, but it must be effected with a knowledge of what can be practically done mechanically and at the same time secure the greatest efficiency of the worker. A carrier medium should be employed wherever a mechanism can take over the work of human hands economically. A man can exert but the fraction of a mechanical horsepower at so many dollars per day, but a large tonnage can be moved by a few mechanical horsepower-hours at an expenditure of but a few cents. There is a further advantage in the steady, untiring service of mechanical equipment and its absolute dependability to handle maximum capacity continuously, and economies are effected in proportion to the practical development and use of the carrier medium.

PLAN OF PROCEDURE

Beginning with incoming material, ease and facility of distribution are the prime requisites. It is a matter of mechanics whether this be at the top, bottom, or on a level with manufacture or stocks. Such locations will be dictated by the premises, the nature of the product, facility of movement, and required frequency of transportation as regulated by quantity or designation. Location and arrangement in this case are more important than the devices used for transportation because of their effect on the remainder of the handling problem, as the degree of fluidity of movement is to a considerable extent dependent on these sources.

Proper arrangement will also be of benefit to the stock-accounting system. Visible inventories under the control of a well-organized planning department contribute to general cost reduction in many ways in this phase of a manufacturing business. Planning departments thus control the movement and set the pace.

STOCK MOVEMENTS

The layout of general stocks, immediate stocks, etc. should be so contrived that the inward movement does not conflict with the outward handling. Aisleways should be provided either by separate lanes or in one general direction, in a manner to prevent cross-travel or reciprocation. This system has become almost an art in wholesale establishments, where there is a constant movement out with an almost continuous replenishing of stocks. A vast number of articles may be quickly gathered from many sources, without error in quantity or article, by a simple system of flowing movement coupled with a suitable control system.

These well-established methods may be adapted with some modifications to the handling of raw or semi-processed material in manufacturing plants, as the fundamentals are quite similar.

HANDLING MEANS

Mechanical means for handling may be variously used, depending on the character of material. Inclosed incoming goods may be economically handled on the continuous automatic type of conveyor if sufficient in quantity, likewise distribution to intermediate storages, through processes, inclosures, storage of finished goods, and shipment. The truck type of conveyance at times finds economic use in this work as a general carrier of miscellaneous material to many selective terminal points.

The selection of the type of conveyance is a matter of individual application. The movement unit and frequency will be regulated by the time elements of consumption in quantities on the basis of production. The most suitable carrier can be determined by considering fixed charges against a conveyor system to reach all desired points in comparison with the cost of the labor necessary to accompany the unit truck load plus truck-equipment fixed charges. The required movement in quantity, frequency, and distance will establish the total truck and labor outlay. On comparing these figures the methods to be employed will be self-evident. Examples of computations of earnings and charges are given later in this paper under the heading Economics.

The same line of thought will also dictate general interdepartmental handling and methods best suited to a reduction of costs, particularly where the departments are impossible of relation to each other in the sequence of operation because of peculiarities of premises or product.

PLAN OF MOVEMENT THROUGH PROCESS

For movement through process, one operation should be made as nearly as possible to the preceding one. This should be the case in effect if not in reality, but the line principle can be achieved in effect by mechanical means, and at times by quite circuitous routes but with practical carrying economy. In the flow of products, if the continuous type of conveyor be utilized, varying distances are of no consequence as the arrivals equal the departures, there being on the line as many units as are necessary for the frequency, speed, and distance. These lines will also act as magazines to provide flexibility in the flow, and will as a rule constitute the "stock in process" storage.

Processing points—machine units, assembly benches, inspection and test stations—should be arranged in their proper order of procedure in the fabrication of the product, concentrated to eliminate

unnecessary motions, and placed in practical proximity to the carrier.

Wherever possible, the carrier should be the sole support of the material moved, serving as the work bench and obviating removal of product from it. This will save much energy and many motions. Where a number of operators can be linked in a line, a stream-like movement can be provided that will coördinate the work of many with little supervision, and set a desirable pace.

Level sections or lines of roller conveyor are used with success for all sorts of processes, inspection, testing, repairs, packing, and shipping; operators stopping the article sufficiently long enough for the task, and then advancing it by hand to the next operator; or where there is group operation, such groups may be stationed parallel to a line and the product deflected over a short lateral for process, thence to an outgoing parallel line to the next group, and by this offset route be adapted to group operation with any required variance in group size.

In movement to or through process some products lend themselves well to a scheme of stock control at intermediate points, or at a point of receiving from a contributing department. There the goods are arranged for distribution to various process points, depending on the character of operation, or the variety of articles, styles, or sizes. Sometimes the work of distribution is sufficiently simple to permit of a circulatory or loop system where the article may return, if not removed on first trip, as an untimed source of supply.

Another method is to predetermine the destination by devices on the carrier in a manner to automatically discharge the movement at the required point. Also by the use of any number of tributary lines a required number of routes may be acquired to secure transportation to handle many diversified products on the same principle of movement and control. In the manufacture of diversified products some may be adaptable to complete mechanical handling and others only partly so. At any rate, each is considered on its merits, with the result that the various products that have more or less complete areas of process, with their attendent systems, are linked up with the necessary sources of supply and collection in the outward direction, and in a manner to suit conditions.

In case of isolation of department for temperatures, air conditioning, abrasive work, dust, and the possible necessary return for reprocess in these areas, carrier lines may be selectively rerouted or overhead return lines used to secure the complete cycle of movement.

There are so many ways in which the flow of products can be and is arranged as prescribed by local requirements that space is not sufficient for a description of a comprehensive assortment of plans. The few applications described above, however, will serve to indicate the possibilities in employment of means to reduce costs and to stimulate interest in a subject that is comparatively obscure.

Packing and Shipping Rooms—Storage

A well-developed use for carrier equipment is in packing departments linked with shipping rooms and warehouses or storage areas. Here the conveyor system has a splendid adaptation if well designed in plan. Products may be sent in in regulated quantities or as produced and a continuous flow be obtained to the process of inclosure. Large labor savings can be made in packing and shipping departments by the use of handling equipment.

The conventional method is to place operators in a line with the flow of products, and have the latter picked up or deflected to the various areas for enclosure. Products that are boxed, crated, or carton-covered are frequently so enclosed on the conveyor or in a manner similar to many of the process operations previously described.

Where there must be checking operations, the goods come to the checkers in the same manner, adjacent to packers, who, when they have completed the task, send out the packed unit on an adjacent conveyor line to storage or to the shipping room. The empty packing containers can be supplied to the packers over conveyor lines parallel to them, as a continuous source of supply. From packing rooms tributary lines may lead to storage areas, and with a circulatory or loop arrangement of conveyors in the storage areas, serve to load in and out of storage in one direction. The shipping room may have carrier equipment connected directly with the packing room or storage, to facilitate the work in practically continuous movement out to trucks or common carriers.

These systems may be extended directly into railway cars at selective points, or by a traveling tripper-chute scheme or deflection find any desired car in a coupled string on the track.

The Movement Unit

The quantitative movement unit may be any suitable quantity of completed sub-assemblies of mechanical goods, parts, articles, or packages, singly or in containers, depending on size and adaptability to standard carrier equipment, and, if possible, corresponding to cost-accounting units. Packages may be handled in a string of singles, or in units in numbers suitable for carrier containers. For conveyor application, usually a maximum size 36 in. long, 24 in. wide, and 20 in. high is desirable for about 85 per cent of the average run of materials and manufactured goods. For miscellaneous packages of light and bulky character much wider paths of travel can be employed. This refers to a more or less continuous stream of materials, namely, cases, crates, bags, and packages, or trays and containers in circulatory movement.

However, carrier equipment may be designed especially for larger

FIG. 8 LOADING OUT FINISHED PRODUCT TO RAILWAY CARS BY MEANS OF A CONVEYOR TRANSIT LINE THAT FEEDS THE PRODUCT AS REQUIRED

(Here is slow-moving storage proceeding to car with three operators engaged in the task of shipping a great quantity of goods.)

and heavier articles. The proportion of height to length is relatively unimportant if the product is carried substantially on a level system (not floor-to-floor movement). For instance, articles such as kitchen cabinets, furniture, entire mechanical assemblies, or similar products having heights out of proportion to their other dimensions, may be safely transported in an upright position on an automatic conveyor system through various processes to completion, packing, and shipping, on light grades or levels. Packages having length as the greatest dimension can be handled effectively in almost any chosen way.

Application of Conveying Appliances

In the use of automatic conveyors, advantage should be taken of the principle of gravitation wherever it is suited to the product. For unyielding packages, or articles with one smooth plane surface for contact, rollers may be employed in lines or spirals. However, a large number of articles traveling per minute may cause acceleration to the point of "racing," and at times power control is necessary, particularly in the case of mechanical products or fragile goods. Most types of average-sized packages may be handled on friction slides or chutes, or in friction spirals, where there is not too great a diversity of material in frictional contact. Metal, wood, fabric, and paper have different coefficients of friction, and for any given friction grade proper speed control will not be possible for all coefficients. Belt conveyors may be employed for a general run of products that will not injure the belts, and where high speed and high-capacity movement are desired. This has reference to normal

weights and sizes of packages, and belts may run vertical, inclined, or level, with proper accessories.

Belt conveyors running at slow speeds through process also often prove the best carrier medium for all sorts of containers and articles. They are not always limited to size and weight. In one known case, fabric belts in double line carry automobiles in the final assembly. It is a matter of application and the type of carrier—a mechanical question.

In the employment of conveyors using chain there are countless applications. The various manufacturers of these devices issue catalogs that are complete to the point of being textbooks, and no comment is necessary beyond the statement that sizes, weights, and materials unsuited to other types of equipment can in nearly every case be handled on some type of chain equipment. Belt- and chain-conveyor equipment for continuous transportation can be suitably used for movement down, up, and horizontally.

The electric truck is a good distance-hauling medium for the periodic movement of the unit heavy load, and serves its purpose

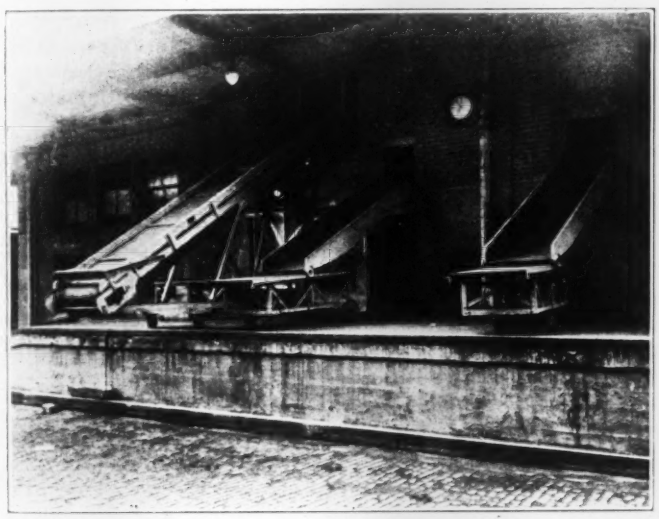

FIG. 9 CHUTES FOR LOADING OUT BY GRAVITATION TO TRUCKS AND AN INCOMING PORTABLE CONVEYOR

(Small extension truck chutes support the lower ends of the chutes, and are used for extension into motor trucks at beginning of loading, and gradually withdrawn as truck is loaded, thus preventing a carry from mouth of chute to head end of truck. A conveyor system inside distributes to the various outlet chutes. When not in use, the chutes are folded away in the wall recesses for sidewalk clearway.)

well for traffic between buildings far apart and in moving material of a character not adapted to the fixed conveyor. Economics dictate its use for widespread distribution to many points, great variety of materials, or where the movement is too light to support the larger fixed equipment.

There are many cases where a number of methods and types of carriers are required. Tiering machines, hoists, pilers, portable conveyors, overhead-trolley systems, electric and hand lift trucks are frequently employed to advantage as auxiliaries to the main system, or may be the sole handling medium. Any discussion of their relative merits would be too involved for the space here available.

The continuous type of conveyor is a good investment when used overhead for through routes, or to permit passage or use of floor space under it, as it utilizes unproductive space above and is not burdened with a floor-space charge. Also it may constitute a feed or storage line, thus reducing storage area. There are in existence many such systems of considerable magnitude that carry the entire output almost exclusively under the ceilings, receiving through floors at selective points above them, and discharging by many routes to many terminal points. Systems designed in this manner serve for the outward handling of finished goods in bags, cartons, or boxes without requiring any extra space for the equipment.

Mechanical-handling equipment has found an extraordinarily economical application in the production of goods involving high temperatures in their manufacture, enabling operators to be at remote points and safe from the effect of the heat. Ceramics, glass products, and the vast number of heat applications to metals when mechanically handled make for a variety of benefits. The same is true of spraying and immersion, both for coatings and chemical operations, where these processes are automatically effected on conveyors. The handling of glassware, electric lamps, and other fragile goods can be done with the minimum of shock or injury by mechanical means.

Losses by breakage and high labor cost are big items in the handling of freight and cargoes, and conveyors are rapidly being developed for these purposes, particularly for classified small packages, which represent the bulk of miscellaneous goods.

POWER CONTROL

Automatic conveyor systems having many selective routes may be provided with the added facility of centralized and remote electrical control for general operation at a desired point, in order that their functions may be governed by a loading superintendent, traffic supervisor, or a planning department. This brings about a further reduction in maintenance and operation costs, synchronizes the system with production, and provides a means of securing the maximum in capacity. In some cases such control is a practical necessity where the system covers a large area.

STANDARDIZATION

Any system employed in mechanical handling should be well standardized throughout the plant, for the reason that it is not usually left intact for a long period of time. Changes in product, particularly in diversified lines, expansion to include increased business, or correcting of errors in judgment developed by use, often necessitates rearrangement, and the equipment should be designed with such a possibility in view, and in a way to permit changes in relocating, lengthening, shortening, etc. so that the same equipment may be employed without discarding parts of it because of unsuitability.

ECONOMICS

After a plan for arrangement and operation has been plotted out and the requisite equipment to carry the production has been

TABLE 1 COMPUTATION OF EARNINGS OF A MECHANICAL-HANDLING SYSTEM

FIXED CHARGES		
1 Depreciation at 10 per cent	$ 5,200	
2 Interest on investment at 6 per cent	3,120	
3 Power required (30 hp. equipped) for 300 days of 8 hours each, total, 54,000 kw-hr., at $0.03	1,620	
4 Operation (3 operators 300 days of 8 hours each), 7200 hours at $0.60	4,320	
5 Maintenance (lubrication, repairs, and incidentals)	780	
6 Insurance at $0.60 per $100	312	
	$15,352	
EARNINGS		
7 Reduction in direct labor (unskilled), 15 men 300 days of 8 hours each, total 36,000 hours at $0.60 per hour	21,600	
8 Reduction in direct labor (skilled), 3 men, same period, total 7200 hours at $0.75 per hour	5,300	
9 Increased production of skilled labor involved (time study, affecting 115 men = 3 per cent), total, 8280 hours average $0.75 per hour	6,210	
10 Reduction in floor space (based on distribution of total production at factory cost over total plant area of 200,000 sq. ft.; production value of $10 per sq. ft., earning at 10 per cent or $1 per sq. ft.) total, 18,000 sq. ft. at $1	18,000	
11 Time losses in delays, viz: elevator waits, congestion, and other causes (time study), total, 8200 hours at $0.60 per hour	4,920	
12 Reduction of stock in process inventory = $75,000, at 6 per cent	4,500	
13 Reduction in manufacturing period, permitting increase of production equivalent to 7.8 per cent of annual output, or total of $156,000, when computed on the basis of $1 earning per sq. ft. as per item 10, is equivalent to additional floor space saved in extent of 15,600 sq. ft., or a total earning of	15,600	
Total Earnings		$76,120
Less Fixed Charges		15,352
Net Return		$60,768

Or 116.85 per cent on investment.

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fitted to it, the total approximate cost for changes, fixtures, equipment, etc. should be estimated in order to provide a working basis for assigning fixed charges. This total should carry a normal item for depreciation of 10 per cent, interest charges of 6 per cent, insurance costs, the cost of power required at prevailing rates, wages necessary for operation of system as an operation charge, and an item of from 1 to $1^1/_2$ per cent for maintenance, repairs, lubricants, and incidental items. As an offset to this total, any or all of the following may be classed as benefits: direct labor reduction in handling, time losses waiting for materials or other delays, productive earning of floor space saved (approximately 10 per cent of the value of products at factory cost, apportioned on the basis of the total output to the total floor space), 6 per cent of inventory reduced, time losses in delays (elevator waits, congestion, and other causes), and the monetary value of a more rapid turnover or shorter manufacturing period. The difference between these two totals represents the possible earnings on the project.

An example of the computation of earnings in the case of a plant producing goods amounting at factory cost to approximately $2,000,000 annually, is given in Table 1. The initial cost of installing a mechanical-handling system complete is assumed to be $52.000

A matter of frequent controversy that may be commented on in conclusion is the question of overhead on labor or other items in an economic analysis of plant operation. Because of the broad blanketing of overhead, ordinary reductions in direct labor will not reduce it because it is still there, and serve only to increase the percentage figure slightly. By the same measure overhead need not be charged on items under the head of fixed charges.

The example given in tabulated form is an average one. There have been many cases where the net earnings have been in much greater proportion to investment.

AFTERWORD

Mechanical handling of some type is applicable to practically every type of industry, but must be varied according to size, product, and characteristics of buildings, and for various other reasons. In consequence it can readily be seen that any attempt to cover the ground in detail would easily fill a volume, and the foregoing is therefore to be considered only as an effort in suggestion, accompanied with a few examples and statements of results.

If manufacturers will focus attention on their movement problems with the same zeal that they run down economies in processing, accounting methods, and sales, they will tap a great mine of profits.

The problems presented are often intricate, but there are available capable engineers who specialize exclusively in this work, and the same progress that has characterized automatic machine processing may be confidently predicted in connection with materials handling.

Scientifically Developed Shipping Containers

Yearly Cost of Packing and Shipping Commodities—Classification of Commodities—Standardization of Lumber for Containers—Wooden Boxes, Crates, Barrels, and Casks—Tests on Containers—Loading and Bracing Containers in Cars

By C. M. BONNELL, JR.,[1] NEW YORK, N. Y.

THE development of a shipping container according to engineering principles is just as much a part of the production and marketing of any commodity as is the commodity itself. Many manufacturers whose products are the direct outcome of applied engineering principles have given very little consideration to this. Without proper containers for shipping, however, a product generally cannot be delivered intact to the consignee, and unless it is delivered intact, the time and money spent in making it have been wasted. More and more manufacturers have realized this in recent years, and many are now using just as much skill and knowledge in the shipping of their products as in the manufacture of them.

Yearly Cost of Packing and Shipping

To estimate the exact total yearly cost of packing and shipping commodities would be difficult. Information that is available will give some idea as to what this enormous sum amounts to. In 1922 and 1923 a survey made by the Freight Container Bureau showed that an average of 182 pieces of freight in containers are placed in less-than-carload merchandise cars. The Car Service Division of the American Railway Association reported that during 1923, 11,877,812 cars of less-than-carload merchandise were loaded. Multiplying the two figures we obtain 2,161,761,784 containers, a figure which is quite astounding but which does not begin to represent the total number used in that year. Add to this approximately 1,535,000,000, which is the estimated number of containers moving in carload traffic during the same period, together with the containers that are used for parcel post and express, and the total number reaches well around 5,000,000,000. Data were obtained in the Freight Container Bureau survey which showed about 90 per cent of the containers moving by freight were new ones, and that about 40 per cent of the total were made of wood in the form of wooden boxes, crates, pails, kits, tubs, barrels and kegs.

The Bureau of Census of the Department of Commerce reports in the 1921 Census of Manufactures that the total value of products of manufacturers whose principal products were containers amounted to $367,786,417. When the containers made by other manufacturers whose principal products are something else are considered, the total value of all containers manufactured during that year would amount to considerably more than this. It is estimated that the state of Florida alone uses 20,000,000 orange boxes yearly and the Northwest about 13,500,000 apple boxes yearly. There are roughly 325,000,000 pairs of boots and shoes made annually in the United States, and of these about 78,000,000 pairs are shipped in wooden boxes. Allowing three dozen pairs per outside container at an average of eighty cents per box, the annual bill for wooden boxes in this industry alone would amount to $1,740,000.

Many more figures such as these are available, but those that have been quoted will show that a saving of only a fraction of a cent per container would effect a very large total yearly saving to an individual shipper using a large number of containers per year.

In addition to the actual cost of containers, the shipping bill is further augmented by the cost of labels, wrappers, cartons, etc., and labor required to prepare the articles for shipment, which latter probably amounts to as much as the initial cost of the container alone.

The shipping bill paid to the railroads and other transportation agencies for carrying commodities to their destination is huge, and when this is considered we begin to realize the magnitude of this proposition.

The gross operating revenue of the railroads of the United States during 1924 amounted to somewhat more than $4,000,000,000, and this does not include the amount received from express companies. To this total we must also add the amounts paid to motor-trucking and water-transport companies and for team delivery of freight.

In return for this enormous freight bill the railroads in 1924 handled 48,527,227 carloads of freight. During the same time the carriers paid out a total of $48,262,543 in freight claims filed against them for loss, damage, delay, etc. in transit. The survey on freight containers made by the Freight Container Bureau in 1923 developed that the cost to the railroads for one year (1922) through the use of poor and inefficient freight containers in less-than-carload traffic was as follows:

Recooperage costs	$ 4,539,700
Claim payments for causes involving containers	5,997,536
Claim machinery for handling above payments	964,034
Total	$11,501,270

This represents less-than-carload shipments alone and the same figures for carload shipments would probably double the amount; thus it can be seen that the loss to the railroads on account of poor and inefficient containers is a very large sum. The loss to the shippers for damage where the carrier was not at fault and for refunds to consignees in the case of damage not great enough to file claims for, as well as the loss to consignee of customers that are dissatisfied because goods arrive in damaged condition, represent, an economic waste which if it were possible to figure in dollars would possibly represent a loss equally as great as that borne by the carriers.

The Freight Container Bureau

The Freight Container Bureau of the American Railway Association was organized in 1921 for the purpose of making studies of shipping containers and to prepare recommended engineering specifications for these that would be efficient and economical to use. Studies of the packing of commodities are assigned to trained engineers, one commodity at a time to an engineer. Field investigations are made at plants where the containers are manufactured and used. Visits are made to railroad stations, transfer stations, and destination points to observe the containers in transit. The consignees are also interviewed and containers are inspected at destination, so that a complete picture of the packing and handling of the commodity is obtained.

The work, then, is done in coöperation with manufacturers, shippers, consignees, container manufacturers, and transportation interests. When the field investigations are completed, tentative specifications are prepared and distributed to those most interested for their criticisms and suggestions. When the specifications have been revised so that the provisions meet with the approval of the best-informed and principal manufacturers that the engineer has conferred with, they are issued in circular form and distributed to every one who is interested. In this form they are tentative and recommendatory, and represent methods of packing that will be sufficient to carry the products safely to destination with ordinary handling.

Classification of Commodities

The number of investigations that have been made by the Bureau is comparatively small due to the amount of time required to complete each one and to the small scale on which the Bureau has been operating. Commodities are classed for claim purposes by the American Railway Association into thirty-three different groups,

[1] Engineer, Freight Container Bureau. Mem. A.S.M.E.

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each group covering one commodity or class of commodities for which various types of containers are used for shipping. Many of these commodity groups can be further subdivided into individual commodities all coming under the same general head, so that the work on any particular group may cover a considerable period of time. For instance, one of these commodity groups is No. 18, Glass, Glassware, and Crockery, which, when subdivided, is found to include, plate glass, window glass, glass tableware, glass tumblers, earthenware, chinaware, sanitary pottery, glass bottles, etc., each of which would require an individual investigation because it is made by a different group of manufacturers and methods vary in each group.

The commodities as classified by the American Railway Association may also be classed in another manner according to the susceptibility of damage as follows:

I Products in return containers
II Products easily damaged and of high value
III Products not easily damaged and of high value
IV Products of low value.

In Class I are oil in wooden or steel barrels, empty tin cans moving to a cannery, etc. While some commodities in this class may be easily damaged, most of the containers are very substantial and do not enter appreciably into the claim account. In Class II are such articles as glassware, chinaware, furniture, stoves, machinery, etc., while Class III includes package groceries, shoes, tobacco, etc., and Class IV cheap glassware and earthenware, commodities in bulk in containers, etc.

Various shippers of these goods may also be classed into the following different groups according to the class of containers and the care used to insure their products reaching destination safely:

1 Those shippers that insist upon efficiency in their containers without limiting this efficiency to the minimum requirements of the tariffs in effect
2 Those that limit the efficiency to the minimum requirements of the tariffs
3 Those that consider only the cost and are willing to sacrifice efficiency and disregard minimum requirements when acceptance of the freight by a carrier can be secured.

In the following discussion the principal types of wooden containers used for shipping will be described, and particulars will be given of some of the work that has been done in the past few years to improve them.

Wooden Boxes

First of all, the lumber to be used in the construction of wooden boxes is important. Some species are harder and denser than others, hold the nails better, and will not break in shearing or bending as easily as some of the less dense woods. The Forest Products Laboratory of the United States Department of Agriculture has divided the various species that are used for boxing and crating into four groups according to nail-holding power, physical strength, and general considerations necessary in making a good box. Many are not conversant with this grouping, but it has been in use for several years with very good results. The grouping is as follows:

Group 1

Alpine Fir	Cucumber	Sugar Pine
Aspen	Cypress	Western Yellow Pine
Balsam Fir	Jack Pine	White Fir
Basswood	Lodgepole Pine	White Pine
Buckeye	Magnolia	Willow
Butternut	Noble Fir	Yellow Poplar
Cedar	Norway Pine	
Chestnut	Redwood	
Cottonwood	Spruce	

Group 2

Douglas Fir	Southern Yellow Pine	North Carolina Pine
Hemlock		
Larch		

Group 3

Black Ash	Pumpkin Ash	Tupelo
Black Gum	Red Gum	White Elm
Maple (soft or silver)	Sap Gum	
	Sycamore	

Group 4

Beech	Maple (hard)	Rock Elm
Birch	Hickory	White Ash
Hackberry	Oak	

Standardization of Lumber. For the purposes of engineering-container specifications the standardization of lumber sizes would aid materially. It is necessary in properly designed and constructed containers to know the exact thickness of material so that proper attention can be given to nailing and to the strength of members. The size of nail that should be used depends largely upon the kind of material and thickness. In the past a commercial 1-in. board might vary from $^3/_4$ in. to $^{15}/_{16}$ in. in thickness, and it was impossible to figure with any degree of accuracy which thickness would be obtained when it was desired to make use of it. The recent standardization activities will eliminate much of this uncertainty.

Moisture Content. The moisture content of lumber for boxes is a very important factor. A box that is made of wet lumber and allowed to dry out will be considerably weaker than one in which lumber of the proper moisture content has been used. Also alternate wetting and drying will reduce the strength of the box each time. The proper moisture content of lumber for boxes is between 10 and 20 per cent, based on the weight of wood after oven-drying to a constant weight. This has been determined by tests made at the Forest Products Laboratory. If the boards are too dry, they break on account of being brittle. If they are too wet, the holding power

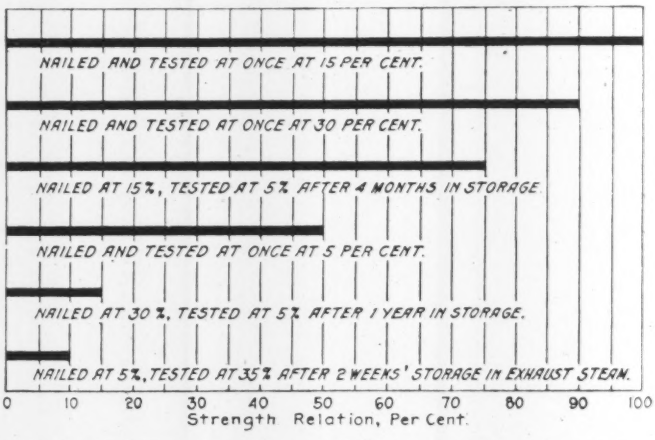
NAILED AND TESTED AT ONCE AT 15 PER CENT.
NAILED AND TESTED AT ONCE AT 30 PER CENT.
NAILED AT 15%, TESTED AT 5% AFTER 4 MONTHS IN STORAGE.
NAILED AND TESTED AT ONCE AT 5 PER CENT.
NAILED AT 30%, TESTED AT 5% AFTER 1 YEAR IN STORAGE.
NAILED AT 5%, TESTED AT 35% AFTER 2 WEEKS' STORAGE IN EXHAUST STEAM.
0 10 20 30 40 50 60 70 80 90 100
Strength Relation, Per Cent.

FIG. 1 CHART SHOWING THE EFFECT OF MOISTURE ON THE STRENGTH OF BOXES

of the nails is impaired when the lumber dries out. Also for some commodities the use of wet box lumber may cause sweating or molding within the container, and this will damage the contents.

Styles. There are various styles of construction in wooden boxes, the principal difference being in the end construction. The styles are designated by numbers as Style No. 1, Style No. 2, etc. Style No. 1 is a plain-end box; Style No. 2 has double-cleated ends, four cleats per end, two horizontal and two vertical; Style No. $2^1/_2$ is a similar box with top and bottom cleats set into vertical cleats, giving additional protection against skewing and weaving in transit. Style No. 3 is the same as Style No. 2 except that cleats are mitered; Style No. 4 has single-cleated ends, two vertical or horizontal cleats across the grain of the wood in the ends; Style No. 5 is the same as Style No. 4 except that the cleats are placed inside of this box and are sometimes made triangular in cross-section; Style No. 5 is a plain locked-corner box.

Fasteners. Most boxes are fastened together with nails, these being the most common fasteners. They serve to hold the parts together and provide rigidity. One of the principal defects in boxes is the nailing. Generally too few nails are used to obtain the full strength of the other material in the box. This is also the condition when nails that are too small are used. Nails too large increase splitting and cause premature failures when subjected to stresses. Proper nailing of boxes demands nails of the right kind, size, and number. Nails should never be driven into the end grain of the wood unless this is unavoidable. The nail-holding

power in the end grain is much less than in the side grain, and the durability and serviceability are correspondingly decreased.

The serviceability of a nailed joint varies with the details of the nails and nailing. The character of the shank surface, the length and diameter of shank, and the size of nail head greatly affect the joint. Many nails in use today for box construction are improperly proportioned. Nails should have reasonably large-diameter heads with proper thickness of head to prevent shearing from the shank. The shank diameter must not be so large that it will cause splitting of the material when driven, but it must be heavy enough so that it will not bend in driving.

A special type of nail known as the box nail is made for use in the box industry. Box nails may be cement-coated, barbed, or plain. The cement or rosin coating increases the friction between the shank and wood, and thereby gives greater holding power. Generally a barbed nail has no greater and sometimes less holding power than a plain nail. Tests have demonstrated this. Plain nails are ordinarily used for cleats because the nails are clinched. Cement-coated box nails have from 20 to 40 per cent greater holding power than plain or barbed nails and are cheaper than either of the others.

The character of the contents of a box greatly affects the design. Some contents contribute strength to the package, while others are fragile and offer no support to the outside container, so that all the strength must be in the box itself.

Other factors affecting the design are weight, size, inside packing, and value of commodity. With all these conditions given, the shipper should be able to design and construct his shipping containers with a reasonable assurance that they will reach destination in good order.

Metal Binding. Extensive study has been made of the metal binding of boxes. This usually permits the use of thinner lumber and renders the box difficult to pilfer. Wire binding or flat metal strapping is used. The wire is fastened around the ends or center of the box by twisting the ends of the wire together with a special tool. The metal strapping is attached in a similar manner and secured by pulling it tight around the box and fastening it by the use of a special machine. Metal strapping may also be applied by nailing.

CRATES

For crate construction the available species of wood most suitable have been classed into two groups. The first of these groups contains those species in groups 1 and 2 for box construction, while the second contains those in groups 3 and 4 as classified for box construction. This grouping, which is based largely on strength considerations as structural members, has been in use for some time and has proved quite satisfactory.

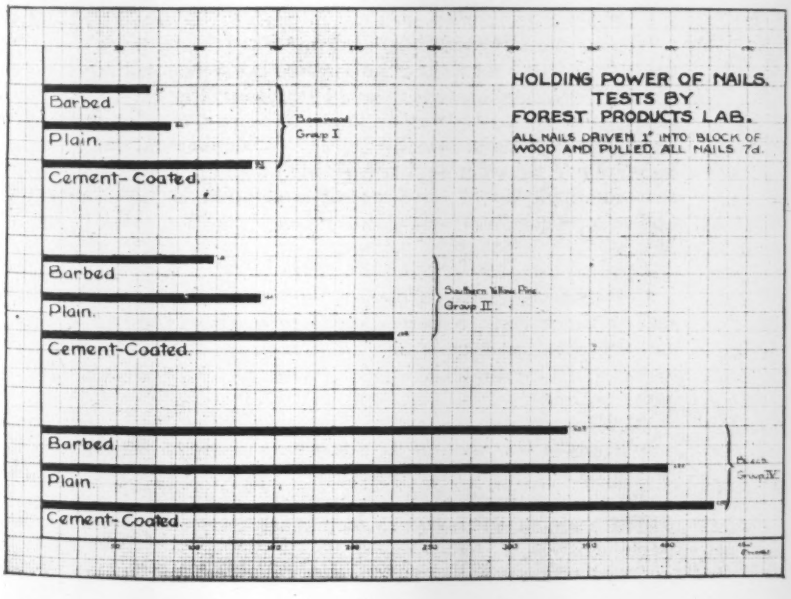

FIG. 2 CHART SHOWING RESULTS OF TESTS ON THE HOLDING POWER OF NAILS

One of the principal faults in crate construction is the arrangement of frame members, all of them in a poorly designed crate being either parallel or perpendicular to one another. This means that the frame members form rectangles, which are not rigid polygons. The triangle is the only rigid polygon, as it cannot change its shape without changing the length of one or more of its sides. This means that diagonal braces must be used in order to obtain the maximum

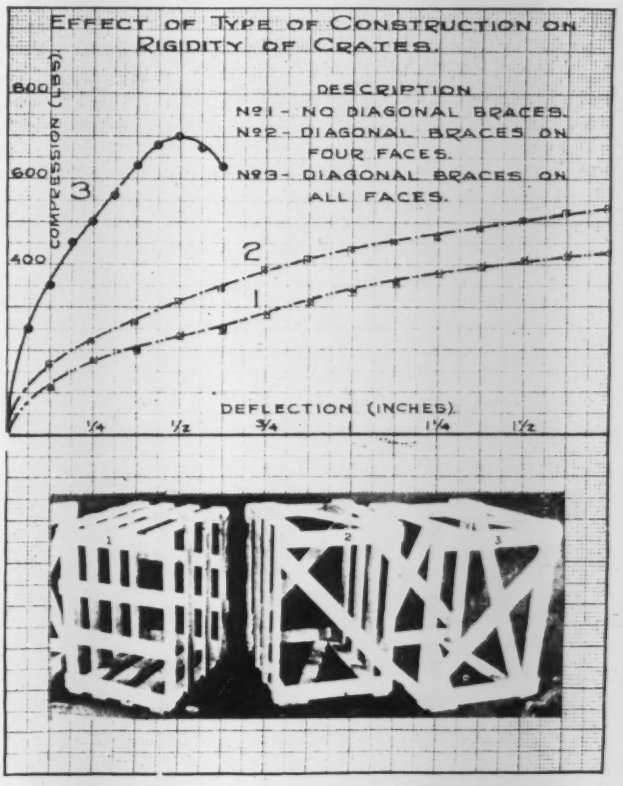

FIG. 3 CHART SHOWING THE EFFECT OF DIAGONAL BRACES ON THE RIGIDITY OF A CRATE

rigidity and strength. The type of construction is similar to that used in steel bridges. Crates made with frame members parallel and perpendicular are usually not rigid and are very easily skewed out of shape, thus allowing the article within to be easily damaged either by rubbing finished surfaces or by breaking fragile parts, because the stress which the crate should carry has been transmitted to the article itself.

Diagonal braces may be used in a number of ways. They may be applied to some or all of the crate faces. Two diagonals per crate face are better than one, and likewise one is better than none. Tests on crates made by an engineer of the Freight Container Bureau showed the following:

1 The maximum load a crate can withstand with diagonals on four faces varies inversely as the ratio of the areas of the faces from which the diagonals were omitted.

2 If diagonals of the same size as frame members are used, the crate will be 15 per cent stronger if the diagonals are nailed to the wide side of the frame member than if nailed to the edge. If the nails are clinched when nailed to the wide side, the advantage is increased 60 per cent.

3 A crate with diagonals nailed to the narrow side of frame members is $3\frac{1}{2}$ times as strong as one without diagonals.

4 Two metal-strap diagonals per face properly nailed give greater strength than a single wooden diagonal. A single metal-strap diagonal is of no value.

The crate corner is also a weak part of the ordinary crate. Several different types are in use, but the three-way or lock-corner type has generally proved to be the best. This type is one in which each of three members forming the corner is nailed to both of the

others, all nails being driven into the side grain. Nailing is just as important with crates as with boxes. Cement-coated nails should be used for reasons stated before.

The moisture content of lumber is also important and should be between 10 and 20 per cent, based in the weight of the wood after oven-drying to a constant weight.

Slack Barrels and Slack Casks

These containers are made of staves cut to certain thicknesses and lengths. Staves are cut wider at the center than at the ends, making the container larger in diameter at the center or bilge than at the ends or heads. They are made of various species of lumber such as red gum, oak, hickory, elm, basswood, poplar, etc. In times past these were hewn out of splits from the logs, but now the logs are cut to the proper lengths, sawed into splits, and then either sawed or knife-cut into staves.

The staves of a barrel are held together by hoops of a definite diameter; if the staves are not properly seasoned, shrinkage in width occurs which reduces the circumference of the barrel; as the hoops do not shrink with the staves, this causes a loose or weak container. The proper moisture content of staves is between 8 and 12 per cent, based upon the weight of the wood after oven-drying to a constant weight. The moisture content should not be less than 8 per cent, as staves will crack and split from being too brittle.

The hoops for barrels may be any of the following: patent wooden, oval bark, flat steel, beaded steel, steel wire. Tests made by the

TABLE 1 COMPARATIVE STRENGTHS OF VARIOUS KINDS OF BARREL HOOPS

	Hoop tension, lb.
Patent wood hoop (elm, 2 nails at joint)	634
Patent wood hoop (elm, 3 nails at joint)	850
Oval bark hoop (3/4 in. wide)	1408
Wire hoop (No. 11 gage, twisted-splice joint)	800
Wire hoop (No. 11 gage, butt-welded joint)	824
Beaded steel hoop (1 1/4 in. x No. 23 gage, riveted joint)	1370
Flat steel hoop (1 in. x No. 20 gage, riveted joint)	1890

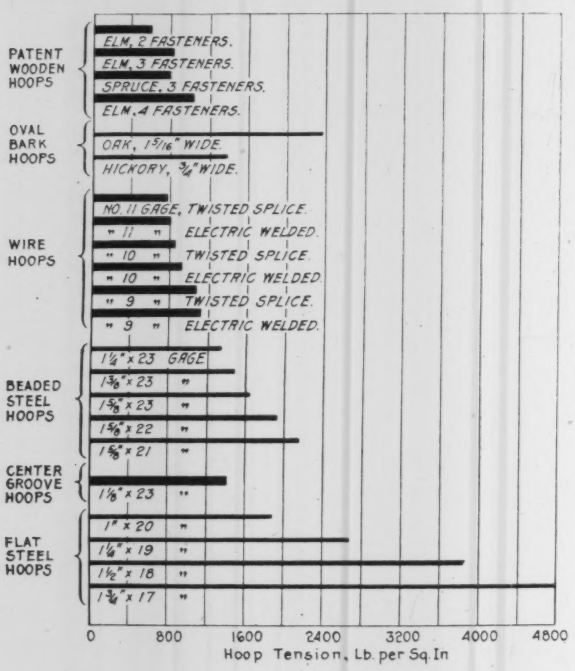

Fig. 4 Chart Showing the Results of Hoop-Tension Tests of Various Types of Hoops

Freight Container Bureau showed the relative strength of the various kinds, and it was found that ordinarily the weak part was at the joint, especially on the patent wood hoop. Hoops may vary in size somewhat according to the size and purpose for which the barrel is to be used. A brief summary showing the comparative strengths as determined in the tests mentioned is given in Table 1.

Other sizes of hoops mentioned were tested and all of the data were prepared and published in a report, a copy of which may be obtained upon request from the Freight Container Bureau. The values as given in Table 1 do not necessarily mean that a hoop having the highest hoop tension is the most desirable one to use for a given purpose, for there are other things such as cost of hoops, adaptability, ease in applying, etc. which must be thoroughly considered before a decision can be made as to what type should be employed.

Aside from hoops and staves, the method and care used in manufacture and assembling have a great influence upon the strength of the container. The ends of staves are notched or crozed to receive the head of the barrel. The croze must not be deep enough to weaken the stave at the head, neither must it be so shallow that

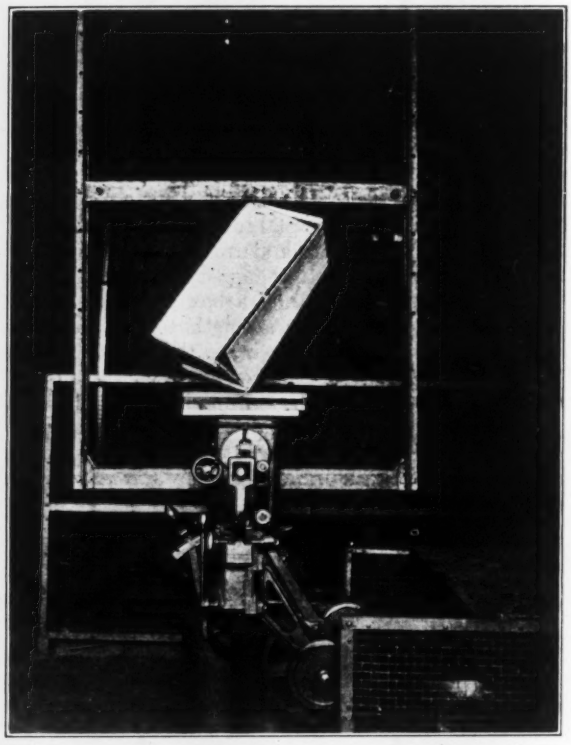

Fig. 5 Compression-on-Corner Test. The Machine Shown in the Rear Is a 14-Ft. Revolving Box-Testing Drum

the head will not hold securely in place. The correct depth is about one-third of the thickness of the stave, 1/8 in. being the minimum. The length of the chime, which is the distance from center of croze to the end of staves, should be about 3/4 in. A short, stubby chime is to be desired rather than a long one, as it is much more rigid and will resist shocks better. When barrels are made by hand, one of the common faults is that hoops are not driven tightly; machine-made barrels have the advantage that the hoops are better and more securely driven.

Tests on Containers

Certain tests have been devised for determining the serviceability of containers and for developing improved ones. Physical determinations of the properties of materials are made, the chief ones being moisture content, breaking strength of wood, and holding power of nails.

Moisture content is readily determined by oven-drying small specimens of the material to a constant weight. The samples are weighed before and after drying.

Breaking-strength tests to determine bending and shearing resistance are made in the usual manner for testing wood or steel specimens.

The holding power of nails is ascertained by driving certain sizes of nails to a determined depth into a block of wood and pulling them out by the use of a tensile testing machine.

The following laboratory tests are made on built-up and packed containers:

1 Diagonal-compression test
2 Drop test
3 Revolving-drum test
4 Vibration test.

Special tests are sometimes made where the standard tests will not suffice. For example, in determining the best packing to be used for eggs it was found that a box of eggs could not be tested in the

revolving box-testing drum because the hazard was too great and the results would not show what was desired. Consequently a special machine similar to the one used for the vibrating test was built and used, and the results obtained by testing therein were satisfactory.

Diagonal-Compression Test. The compression-along-an-edge test, as its name implies, is a steady and constantly increasing pressure (measured in pounds) applied along any edge and with the opposite edge diagonally through the box in a direct line with the pressure exerted. The cornerwise test is applied in the same way to any corner of the package with the opposite corner in a direct line with the pressure. These two tests measure the strength of the box in withstanding any external pressure and to a limited extent

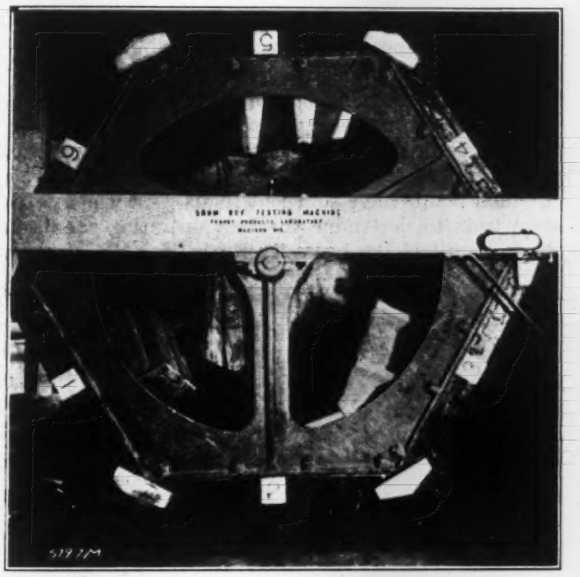

FIG. 6 7-FT. REVOLVING BOX-TESTING DRUM

approximate the hazard of the lower tiers of boxes in a pile. By themselves, however, they are insufficient to determine comparative weaknesses in the various factors that enter into properly balanced construction.

Drop Test. A good test is the drop test, especially for comparing the strength of one box with that of another. In this test the box is packed with actual contents and dropped from a predetermined height directly on the corner. The value of the conclusions in any constructive way is limited, however, because one failure runs so rapidly into another that the observer does not always see the true cause of the weaknesses.

Revolving-Drum Test. The most practical method yet devised for testing packing boxes is the revolving-drum test. The drum is hexagon-shaped and revolves at the rate of one revolution per minute. The box to be tested is packed with the actual contents, as in commercial service, and placed in this drum. In the drum are arranged a series of hazards which cause the box to follow a regular cycle of drops, falling upon sides, top, bottom, ends, edges, corners, and flatwise upon a projection similar to the corner of another box. These drops simulate the hazards of transportation, excepting the heavy static pressure received by a box in the lower tiers of a pile, which latter is determined by means of the compression-on-edge test described above. The drums in use at present are of two sizes, one 14 ft. in diameter and the other 7 ft.

As the box moves on from one drop to the next the observer notes the beginning of the failure at the weakest point in its construction, and follows the development of that weakness until the box entirely fails and spills its contents.

Vibration Test. In this test the empty box or crate is placed upon the table of the testing machine and braced so that it cannot slide about, or so arranged that it can slide back and forth through a certain distance and strike an obstruction at each end. A weight of several hundred pounds is placed on top of the box to simulate the stowing of boxes in a car. The table of the machine is set on rollers and can move back and forth through a certain distance which is variable. This motion is controlled by an eccentric mechanism and the speed of the table may be varied. The test subjects the containers to severe racking and determines their relative rigidity; it simulates the weaving motion that a box is subjected to when placed under a pile of boxes in a moving freight car.

The test is usually more severe than actual shipping conditions, but for obtaining comparative data on containers it is very good, and any lack of rigidity becomes apparent immediately.

LOADING AND BRACING

The loading and bracing of containers in a car usually requires the use of much lumber, and while this may seem foreign to the subject of shipping containers, it is very closely related. Several years ago tests were made at the Pennsylvania Railroad yards at Altoona, Pa., to determine the impact between pieces of loose freight when placed in cars that were shifted about and coupled at various speeds. It was found that at a speed of five miles per hour at the time of impact the force exerted by any container against the end of the car or bracing would be about five pounds for every pound of weight in that container, friction included. It needs no explanation to show that a carload of merchandise weighing 75,000 lb. would require considerable bracing to keep the contents from shifting about in the car. With proper attention to design

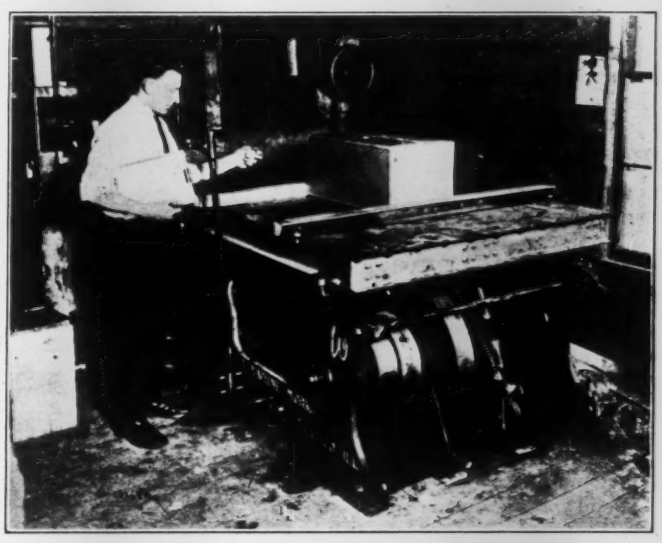

FIG. 7 VIBRATING MACHINE FOR TESTING BOXES

and details of construction it is possible to provide satisfactory bracing at a minimum cost.

In view of the high cost of lumber, materials, and labor, any saving that can be effected by the use of better containers or better methods would be desirable. A few examples of experience obtained by individual shippers through the use of better shipping methods should be of interest.

The General Electric Company and the Edison Lamp Works have installed container-testing machines at their plants, and every type of container that it is possible to test is tested periodically. A trained personnel is maintained to do this work. Figures are not available to show definite savings, but it is safe to assume that this department pays, else they would not have maintained it for so long a time.

The E. S. Evans Company, builders of containers for exporting automobiles, etc., of Detroit, Mich., have effected a saving of $50,000,000 in the last eight years; of this $17,000,000 represents elimination of damages, and the remainder is divided between lumber savings, labor, and lessened freight charges on lighter containers.

The Brown & Sharpe Company of Providence, R. I., have revised their packing according to modern methods and have succeeded in reducing crating costs on machinery more than 20 per cent, on parts, etc. over 25 per cent, and on labor 6 per cent. Their present annual saving runs into thousands of dollars.

The losses on account of insecure containers are still very great. Will better packing benefit your business? If so, what are you doing to bring about this condition?

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The Electric Molder

An Analysis of Its Power Requirements and Production Possibilities in Woodworking, and a Comparison with the Belt-Driven Molder

By ANDREW JENSEN, JR.,[1] MARQUETTE, MICH.

THE PURPOSE of this paper is to present a few facts and observations regarding the production possibilities and the power requirements of two well-known and widely discussed types of molding machines, the belt-driven molder, and the electric molder.

The belt-driven molder has long been regarded as a highly efficient and reliable machine, being generally very well built; and in point of service and reliability it has given the woodworking industry good satisfaction. A few years ago a new type of molder was introduced to the industry, namely, the electric molder, Fig. 1, with direct-connected motor-driven cutter heads and motor-driven feeds.

This machine was claimed to be superior in many respects to the belt-driven machine, and in many cases actually replaced the latter. But in spite of improvements, its introduction into the woodworking industry has been slow and the resistance thereto great. In fact, many managers and superintendents of large establishments are today actually opposed to the introduction of electric molders.

Lack of Data on Speeds and Feeds in Woodworking

This slow response in the woodworking industry to new methods and ideas can be laid directly, it is believed, to the lack of knowledge of the proper speeds, feeds, and power requirements for the various woodworking operations.

The metal-working industry has benefited greatly from the development of high-speed tool steels and from the knowledge of the proper feeds and speeds to use for the best results. Metal-working machines are designed and guaranteed to operate at certain speeds and with a definite power consumption. The purchasing agent or manager has but to refer to a handbook to ascertain whether the guarantees of the machine correspond with standard power requirements and speeds, and the superintendent can likewise refer to the handbook and check the speeds and feeds he is using with standards that are reliable and have been proved by innumerable tests to be both scientific and practical.

The woodworker, however, has but meager and unreliable information by which to gage his results, and is necessarily overcautious of the elaborate statements of manufacturers and their salesmen regarding the possibilities of their machines. Since the only criteria he possesses are those which filter in from other woodworking concerns, he prefers to keep the machines and methods which have proved more or less satisfactory to him. Thus both he and the seller are losers.

The author has studied the situation sufficiently to establish to his own satisfaction the claim that there is actually enough difference in the microstructure of birch as it passes through the various stages of moisture content, and sufficient difference in the power requirements in ripping at these stages to warrant the use of different types of tooth design and different grades of material in the construction of the saw. Nevertheless, in the woodworking industry, if a mill which for years had been running nothing but hardwoods should suddenly switch over and run nothing but soft pine, the same equipment, practically the same feeds, and surely the same speeds would undoubtedly be used.

There can be little wonder that the woodworker, having stumbled successfully on to a satisfactory machine speed and feed, is slow to try any new idea or method, as it means floundering around until he hits again upon satisfactory operating conditions. Then, too, few individual plants have sufficient funds with which to carry on exhaustive tests to determine the best operating conditions, and still less do they feel the necessity for so doing.

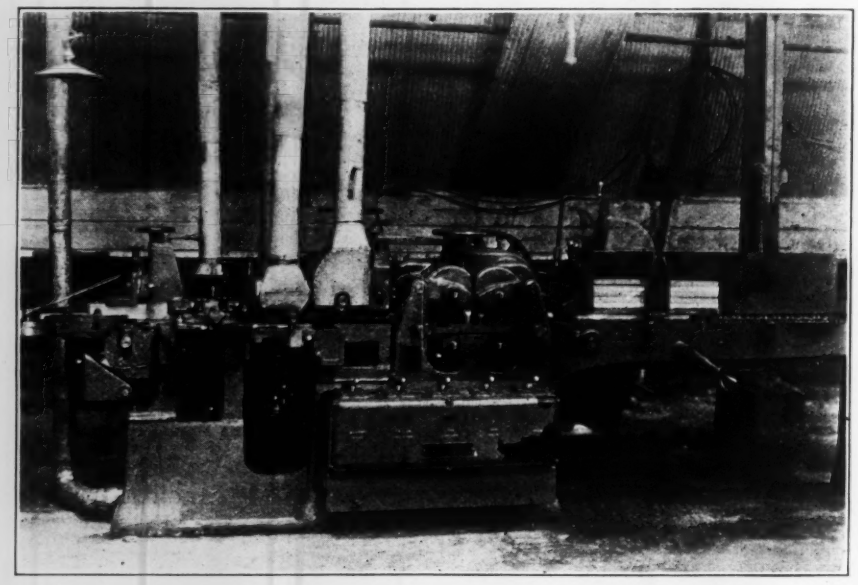

Fig. 1 A Typical Electric Molder

Comparative Tests of Molders

Having read the current literature on the subject of electric molders vs. belt-driven molders, and having watched the operation of different molders and listened to the varying reports of operational efficiency, the author decided to go somewhat more deeply into the study and comparison of the two types of molders, and determine by careful testing and analysis just what were their power requirements and production possibilities.

The tests were conducted under the following conditions:

1 The kind of wood used to be the same in all tests, namely, to be hard wood
2 The grade of lumber to be the same
3 The moisture content to be not less than 5 per cent nor more than 9 per cent
4 The depth of the cut to be the same in all comparative tests
5 The revolutions per minute of the cutter heads to be the same.

The variables were to be (*a*) the speed of feeding and (*b*) the perimeter of the cut.

Of course, the quality of the finished product was to remain within a given limit, which, by the way, did not tolerate rough cuttings or torn-out places, and allowed only the slightest trace of knife marks. The product, in fact, was to be a finished one at the conclusion of the operation.

The tests were run continuously, nine hours per day, for about one month, and yielded the results shown in Tables 1 and 2 and in Figs. 2, 3, and 4.

It can readily be seen, then, that, judged solely on a basis of power requirements, the electric molder has a considerable advantage, and this fact may be accepted as generally true of practically all electrically driven direct-connected machines of the revolving-head type.

[1] Industrial Consultant, Piqua Handle & Mfg. Co.

Contributed by the Wood Industries Division for presentation at the Annual Meeting, New York, November 30 to December 4, 1925, of The American Society of Mechanical Engineers. All papers are subject to revision.

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TABLE 1 POWER REQUIRED AND CONSUMED BY ELECTRIC MOLDER

Time required per 100 ft. feed, min.	Feed ft. per min.	Watts required	Watt-hr. consumed per 100 ft.	Horsepower required
(Cut 1/16 to 1/8 in. deep and 5 1/4 in. in perimeter)				
2.50	40	8300	346	11.10
2.00	50	8500	283	11.40
1.66	60	8850	245	11.85
1.43	70	9250	220	12.40
1.25	80	9700	202	13.00
1.11	90	10150	188	13.60
1.00	100	10700	178	14.30
0.91	110	11150	169	14.95
0.83	120	11700	162	15.65
(Cut 1/16 to 1/8 in. deep and 6 1/2 in. in perimeter)				
2.50	40	8300	350	11.25
2.00	50	8700	290	11.65
1.66	60	9050	252	12.10
1.43	70	9500	226	12.70
1.25	80	9950	207	13.32
1.11	90	10400	192	13.95
1.00	100	11000	183	14.73
0.91	110	11600	176	15.50
0.83	120	12200	169	16.45

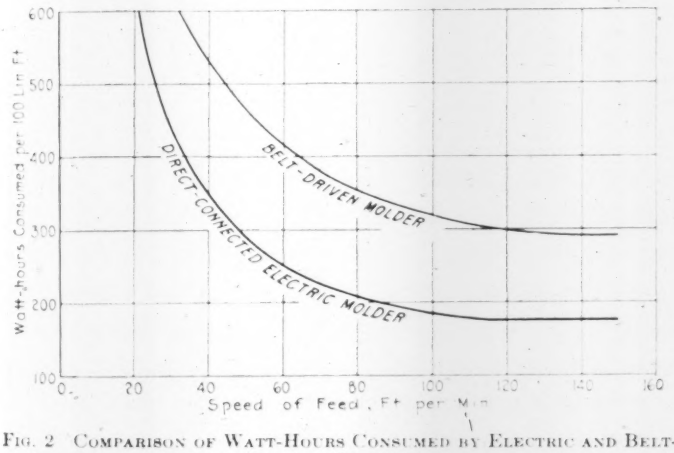

FIG. 2 COMPARISON OF WATT-HOURS CONSUMED BY ELECTRIC AND BELT-DRIVEN MOLDERS

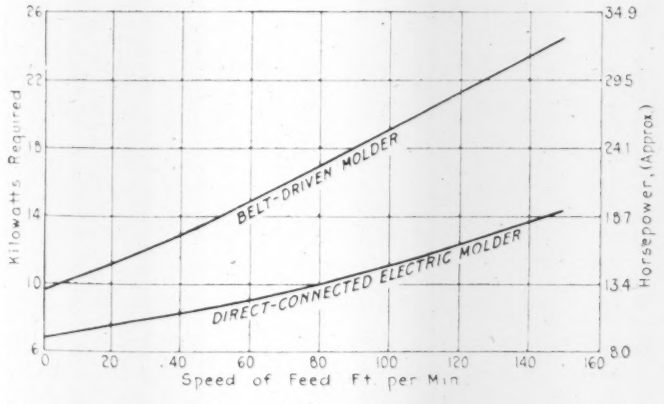

FIG. 3 POWER REQUIRED TO OPERATE MOLDERS AT VARIOUS RATES OF FEED

SAVING IN POWER

In the item of power alone many mills could effect a considerable saving. For example, a small mill buying electric power from an outside source has four belt-driven machines of this type, running on comparatively light work as illustrated in Table 2. The machines run, say, 300 days per year, averaging six hours per day, at a feed of 60 ft. per min. The power required, as reference to the table or chart will show, is approximately 15,000 watts each, or a total of 60,000 watts for the four machines.

This requires 360 kw-hr. per day at, say, 3 cents per kw-hr., which means a power cost per day of $10.80, or $3240 per year.

If this same mill were equipped with electric direct-connected machines, each machine would require approximately 9000 watts, or a total of 36,000 watts, and the daily power consumption would be 216 kw-hr. and would cost $6.48, or $1944 per year. The saving in the cost of power alone would more than pay the interest charges on the additional investment required to buy new and modern machines.

TABLE 2 POWER REQUIRED AND CONSUMED BY BELTED MOLDER

Time required per 100 ft. feed, min.	Feed, ft. per min.	Watts required	Watt-hr. consumed per 100 ft.	Horsepower required
(Cut 1/16 to 1/8 in. deep and 6 1/2 in. in perimeter)				
2.50	40	12800	533	17.15
2.00	50	13900	464	18.60
1.66	60	14900	412	20.00
1.43	70	16000	382	21.40
1.25	80	17000	354	22.75
1.11	90	18000	333	24.10
1.00	100	19000	319	25.15
0.91	110	20100	305	26.90
0.83	120	21200	294	28.40

PRODUCTION POSSIBILITIES

What are the production possibilities of the electric-type machines, is the next question. The following actual experience should answer this question. A certain mill found that it was impossible to fill its orders with its existing equipment, even working twenty-four hours per day, and new electric-type machines were therefore purchased rather under the stress of absolute necessity than because of any real definite knowledge of what the machines could do.

The old machines had, under pressure, been speeded up to a feed of 60 ft. per min., this being the limit at which it was possible to maintain quality on hardwood and, literally, to keep the machines on the floor. A feed of 100 ft. per min. was needed, however, but the various manufacturers would not absolutely guarantee their

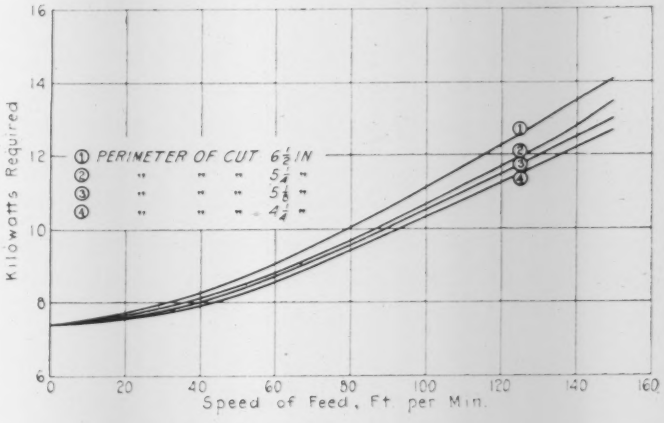

FIG. 4 POWER REQUIREMENTS OF ELECTRIC MOLDER

machines at this speed, even when in some cases the making of the sale hinged upon that one point. Nevertheless, machines were purchased and tried at this speed. The results were not entirely satisfactory for the machines were evidently not designed for this high speed. However, with the aid of the manufacturers, new parts were designed to replace old parts at weak points on the machines, with the result that the high speed was maintained and the quality of the work kept at the customary high standard.

The most satisfactory result of the installation was the reduction in costs per lineal foot of finished product. Formerly the cost of the operation at a feed of 60 ft. per min. had been 55.5 cents per 1000 lineal feet for direct operating labor. The cost on the electric-type machines running at a feed of 100 ft. per min. was only 30.8 cents per 1000 lineal feet.

This startling cost reduction was due largely, of course, to the increased feed, but also to the fact which accurate time study revealed, namely, that the time required for jointing knives, removing heads, making adjustments, and for all ordinary machine delays on the belt-driven machines amounted to 50 per cent of the actual running time, whereas the electric type with their easy adjustment, accessible parts, slip-on heads, and lack of belts required only a 35 per cent delay allowance. The operator's wages were increased about 25 per cent to reward them for the additional speed and effort required.

It may be added here in justice to the electric type that the preceding and subsequent operations were likewise speeded up by the increase in output to a degree not previously possible through a point that had been the "bottle neck" operation. The reduction in costs of these operations, however, will be disregarded in this comparison.

To return to the original example; it has already been shown how an annual saving of $1296 was made in the power costs. Taking the same mill for an example, the comparison of the costs of production are as follows:

Four machines running at a feed of 60 ft. per min. will produce 64,800 lineal feet of product, which at 55.5 cents per thousand feet will cost $35.96 per day; and, figuring three hundred days per year, the annual cost will be $10,788.

Using the figures quoted above for the electric-type machines, this same work could be produced at a cost of 30.8 cents per thousand feet, or $19.96 per day. At this rate the yearly cost would amount to $5988. This would mean a saving of $4800 in direct labor alone, plus a tangible saving of $1296 in power consumption, or a total net saving of $6096 per year. Moreover, to produce this volume of product would require only two electric-type machines, each capable of producing 35,000 feet per day, thus reducing plant requirements to a minimum.

It is not hard to realize why the electric-type machines are fast replacing the old-type belt-driven machines in concerns that have the opportunity to study their problems on a scientific basis; and it is hoped that this paper will aid the numerous smaller concerns that are not fully equipped to study thoroughly their production problems by scientific analysis to make their decisions on these problems, and that it will stimulate further interest in scientific investigation of the many and varied problems of the woodworking industry.

Principles and Advantages of Optical Methods for Measuring Machine Parts

By HENRY F. KURTZ,[1] ROCHESTER, N. Y.

The mechanical engineer, the machine-tool builder, and the machine-tool user are confronted with the necessity of making measurements to a degree of accuracy for which many of the measuring devices hitherto available are inadequate or inconvenient. The properties of light permit the making of four general classes of measuring instruments, some of which may be adapted to their uses. These four classes of optical instruments are outlined in the paper, and measuring instruments already designed for machine-measuring purposes are described and illustrated.

IN MANY CASES light, because of its peculiar properties, may be used for making precise measurements of quantities that are entirely beyond the possibilities of the engineer's usual equipment. The imponderability of light, its frictionlessness, and the fact that it is energy propagated in waves make it useful in many unique ways. The engineer, perhaps owing to the fact that his equipment has been adequate to his needs up to the present, and because it takes considerable effort to overcome the inertia of custom, has been slow to take advantage of these properties of light, even those that have been in successful use in other lines of endeavor for many years. The ever-increasing demand for accuracy at low cost, which is the result of the growth of interchangeability of machine parts, has led to the design, within the last few years, of many optical measuring instruments. The purpose of this paper is to describe these instruments and to point out their advantages, to show what has been done, and to indicate what the future may hold in the way of possibilities.

Types of Optical Apparatus Applicable to Mechanical Engineering Measurements

There are four general types of optical apparatus that have been applied to the uses of the mechanical engineer, though by no means have the possibilities of any one type been exhausted. In fact, the field of application has barely been touched in any of the four directions.

Interference Apparatus. One of the properties of light that was early recognized is the constancy of its wave length for a given color. Another is its property of interference. The combination of these two properties leads to the fact that each band in a group of Newton's rings represents a very definite minute separation between the surfaces forming the rings. This distance is a function of the wave length of the light used, and so the phenomenon permits extremely accurate measurements of small distances. The optician has checked the planeness of surfaces and has compared curvatures and sphericities by means of the interference principle for many decades, but it is only within the past few years that the engineer has used this method in the making and checking of master gages. With careful work he is now able to compare lengths to within observed differences of a few millionths of an inch with certainty. The application of the principle of the interference of light can doubtless be extended to many other phases of mechanical work through the construction of suitable special apparatus, and the optical engineer is ready to coöperate with the mechanical engineer in the development of such apparatus. It would not be delicate, and certainly not complicated, apparatus, and it would require less skill in use than the ordinary micrometer caliper. It is very probable that results would be a hundredfold more accurate and therefore very much more dependable. Interference apparatus should have a wide potential field for usefulness to the engineer.

Optical Levers. A beam of light which may or may not be modified by lenses and prisms into an image-forming bundle of rays, makes an almost ideal lever for the recording of small movements. We all remember how we applied the optical lever in our freshman university days in the study of Hooke's law and in the experimental determination of Young's modulus. But how many of us have used this simple and excellent device since? The beam of light is a nearly perfect lever because it is imponderable and therefore without inertia, because it is perfectly inelastic, because it is perfectly rigid, because it is without friction and therefore without wear; and it is always easily available. Few mechanical elements have such an array of qualities to commend them.

The galvanometer is an example of the widespread application of the optical lever. The mechanical engineer with the help of a cheap cross-hair telescope, a few pieces of mirror, and his own ingenuity could no doubt make excellent use of this device in many places, not only in his laboratory and test room, but in the shop.

Imaging Systems. With the help of lens systems we may control beams of light in such a manner as to form images of objects that we may wish to observe. These lens systems may be designed by a skilled optician to give an image many times larger than the object itself and with a very high degree of geometrical similarity between the image and the object. Obviously the errors of the object are magnified to the same degree, and they become correspondingly easier to observe and to measure. The lens systems may be arranged to present the image for inspection in a plane containing a measuring device, for instance, a scale, cross-hair micrometer, or a standard templet or contour plate, or an accurately made scale drawing of the object under investigation. Such optical apparatus would partake of the nature of the microscope and less frequently the telescope. One of the most serviceable arrangements is that of the projection system, which is coming into use because of the ease in observation, no eyepiece being required, because large magnifications are easily attained, and lastly because a photographic plate may be inserted in the image plane and a permanent and accurate record made of the conditions under investigation.

It might be feared that a record formed on a glass plate which has suffered the vicissitudes of development and drying would not be accurate, that the image on the negative would not be precisely like that impressed upon it by the lens. However, careful investigation of this source of error by the astronomers who use the photographic plate to a great extent for measuring work, has shown that the shift is of the order of one part in 40,000. It is obvious that errors of this order in an image which may be from 10 to several hundred times larger than the object, are negligible to the engineer.

Optical Scale Reading. The microscope together with an eyepiece micrometer makes an admirable device with which to measure small distances. A scale may be made, the least interval of which is equal to the range of measurement of the microscope micrometer, and the two, microscope and scale, incorporated in an instrument for measuring distances with considerable accuracy up to the length of the scale. The scale may be divided on a material for which the temperature coefficient is known and a temperature correction applied, or it may be divided on invar steel which has a sensibly zero coefficient. The scale may be checked by an adequate means at hand or may be checked by the U. S. Bureau of Standards at small cost, and a correction graph or table constructed to correct for accidental or systematic errors of graduation. When this scale is incorporated in a measuring instrument it may be enclosed completely, hermetically if necessary, and observed through a glass window. The scale, then, is completely protected against abrasion or wear of any kind, and is everlasting. The scale, which is the basis of measurement, will last indefinitely and its corrections never change. It will outwear any number of instruments. The instrument itself will be very simple, consisting principally of a slide carrying the scale or the object to be measured, or sometimes both, with possibly an anvil and a contact point directly connected to the slide. In some types a second microscope is used instead of a contact point, relieving the apparatus of errors due to differences in

[1] Optical Engineer, Scientific Bureau, Bausch & Lomb Optical Company.

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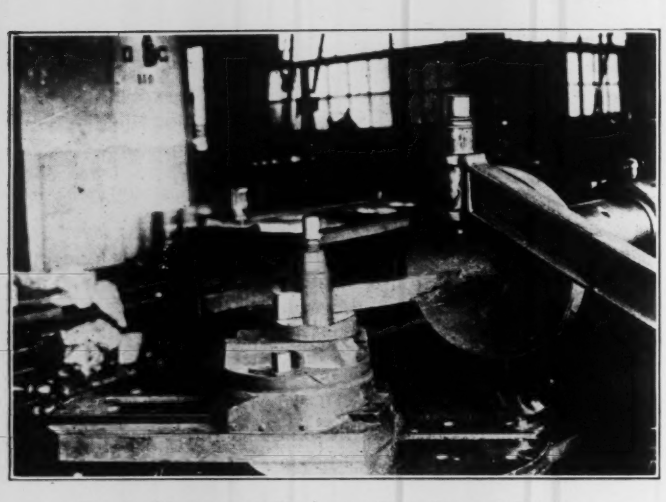

FIG. 1 MICROSCOPE USED AS A TOOL-SETTING DEVICE

(Accuracy of setting, 0.001 in. Setting to this accuracy is made in a few seconds, permitting frequent grinding or replacing of tool.)

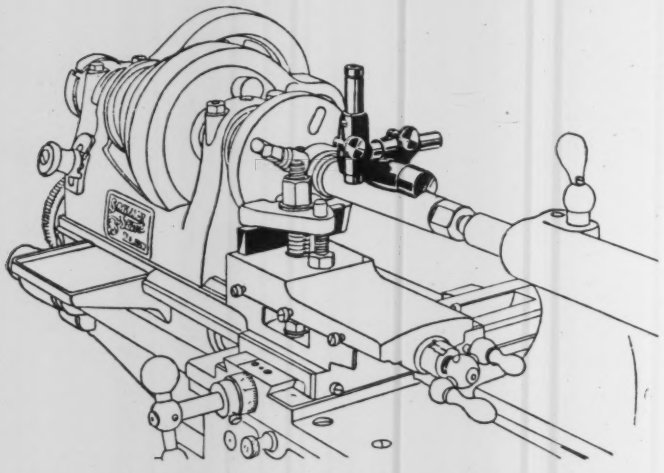

FIG. 2 MICROSCOPE FOR SETTING THREAD-CUTTING TOOL

(Of European manufacture. For setting the tool free from "lean" and for checking angle of tool.)

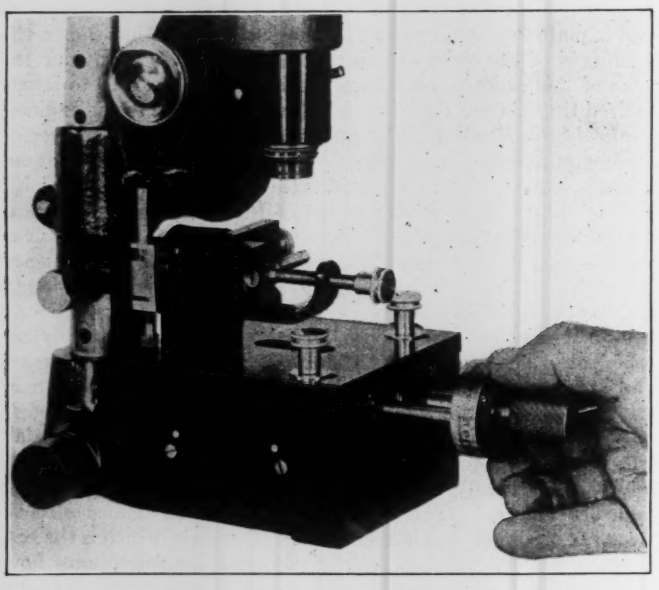

FIG. 5 SPECIAL SCREW-MEASURING STAGE FOR TOOL-MAKERS' MICROSCOPE

(This accessory may be attached to the cross-slide stage shown in Fig. 4. Together with the eyepiece shown in Fig. 6, the flank angle, the angle of "lean," the root diameter, the sharp root diameter, the external diameter, the sharp external diameter, the lead, and the pitch of a screw may be measured to 10 min. of arc for angles and to 0.001 or 0.0001 in. Affords means for critical inspection for finish and workmanship.)

contact pressure. Instruments of these various constructions will be described below.

The advantages of this kind of measurement lie in the fact that the basis of measurement, the scale, is fully protected and is not subject to the slightest wear. It is never touched, and nothing short of complete disaster can affect its accuracy or its life. One can return to the precise unit of measurement after years of

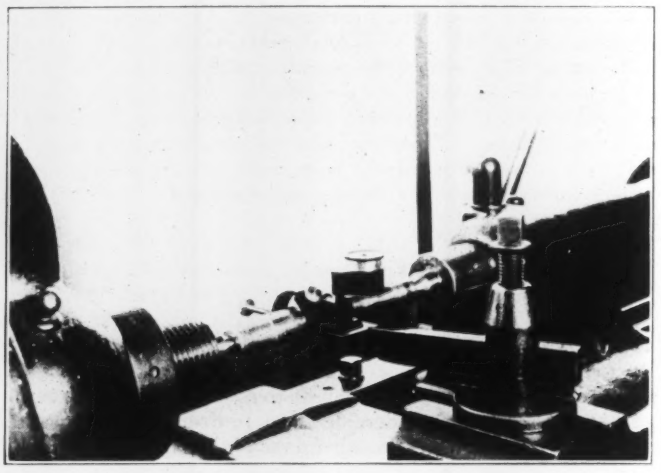

FIG. 3 OPTICAL THREAD-SETTING TOOL

(Of American manufacture. Uses simple magnifier for setting the tool free from "lean," for checking the angle of the tool and setting the tool to the proper height for inside and outside threads.)

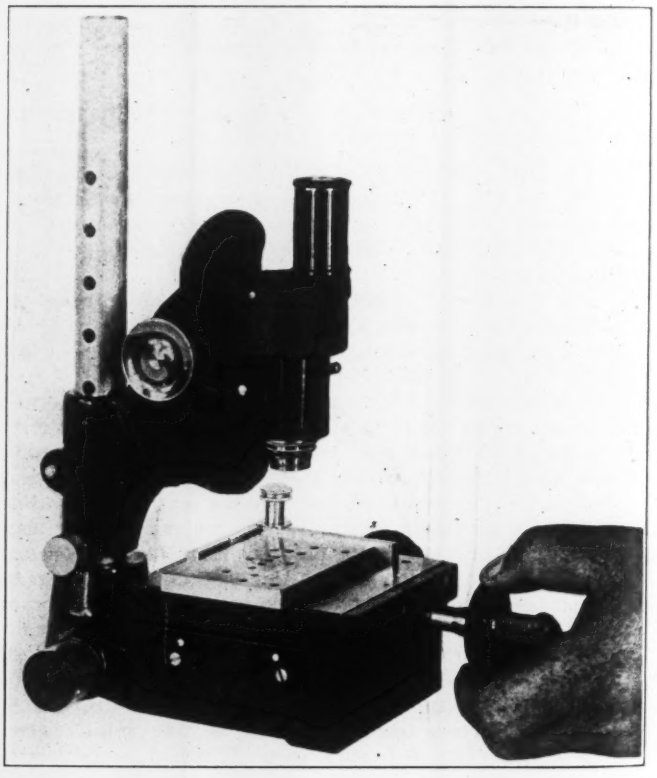

FIG. 4 TOOL-MAKERS' MICROSCOPE

(Erecting microscope, giving the image the same aspect and motion as that of the object. Microscope completely removable from stand for use in other apparatus. Cross-slide stage accurate to either 0.001 in. or 0.0001 in.)

use. Distances of the order of four inches can be measured within 0.00005 in. with certainty.

Apparatus of this type has been used by the astronomer and by the physicist for many years, and it should find wide application in the field of mechanical engineering.

INTERFEROMETRIC APPARATUS—OPTICAL LEVERS

The use of the optical flat together with gage blocks for the comparison of lengths is so well known that it is not necessary to de-

scribe the method and apparatus here. It is mentioned only for completeness. The same principles with the aid of interferometric apparatus can undoubtedly be further applied to the accurate measurement of small dilations and other quantities.

The optical lever consists of a mirror pivoted in such a manner that it will rotate about an axis in the plane of the mirror under the impulse of the motion of the member under investigation. The image of a distant scale formed by this mirror is viewed with the aid of a cross-hair telescope. The length of the pivot arm and the distance from mirror to scale then form the two arms of a lever, the ratio of which is limited only by vibration of the apparatus. This simple device is used extensively by the physicist and the engineer should find many applications for it.

IMAGING SYSTEMS

Fig. 1 is from a photograph showing the use of the microscope for setting the cutting tool in a machine which turns the spherical curves on lens-grinding shells. The microscope containing the cross-hair is movable along the axis of rotation of the shell, on a slide equipped

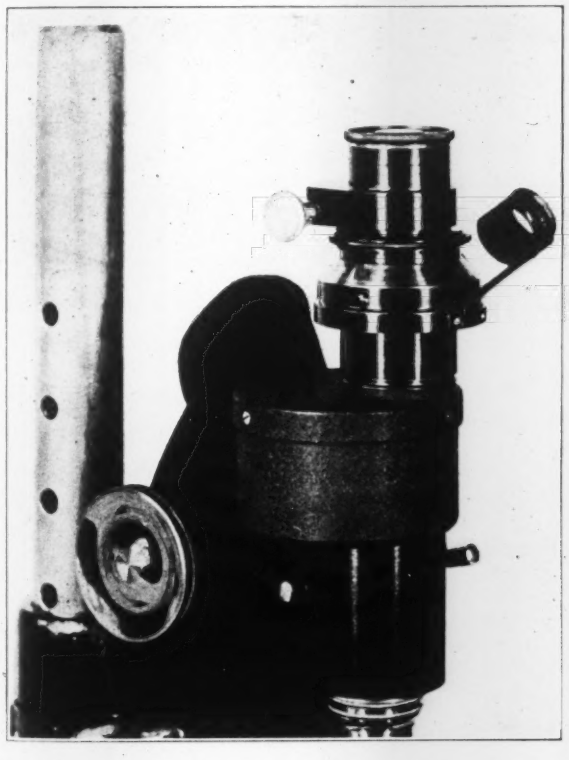

FIG. 6 GONIOMETER EYEPIECE FOR TOOL-MAKERS' MICROSCOPE

(For measuring flank angle and angle of "lean" as well as helix angle of screw, and for making angular measurements on other parts for machinery. Divided to 10 min. of arc; 5 min. estimated easily.)

with a scale. The zero of this scale is adjusted in building to the axis of rotation of the tool holder. The operator then sets the radius of the shell on the slide scale, and clamps his tool in such a position that the image of its cutting edge coincides with the microscope cross-hair. The tool will now be set to cut the desired radius. If the operator wishes to replace or grind the tool, it is the work of only a moment to adjust it to its proper position. During cutting, the microscope is swung back out of the way. The accuracy of setting for this equipment is about 0.001 in. Previous to the adoption of this method the tool could only be set by trial and error, often at the expense of a half hour. There is no doubt that tool setting on many other machines could be equally facilitated with the aid of suitable optical apparatus.

Fig. 2 shows a thread-tool-setting microscope of European make. It consists of a microscope carried by a V-block in such a manner that a templet in the focal plane of the microscope is at the correct tool position for thread cutting. The tool is viewed through the microscope while resting on a cylinder between centers, or while the setting tool is held between centers. The cutting tool is then adjusted to the templet in the focal plane.

Fig. 3 is from a photograph of an American tool for the same purpose. This instrument is provided with a bubble level and means for clamping to the cylinder, usually the piece upon which the thread is to be cut. There is also a templet, but it is in the focus of a simple magnifier instead of in the focal plane of a microscope. When the instrument is clamped to the work cylinder at the horizontal position as indicated by the level, the templet is "at center"

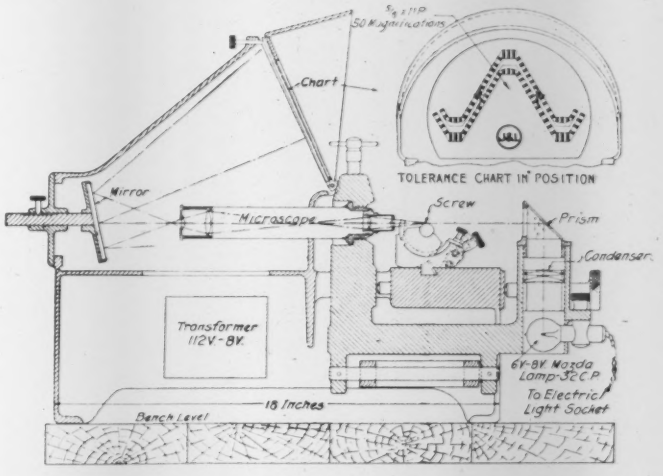

FIG. 7 THREAD COMPARATOR

(Of American manufacture. Permits rapid and dependable comparison of screws in mass production with standard contour having "go" and "no go" limits. Very convenient in manipulation.)

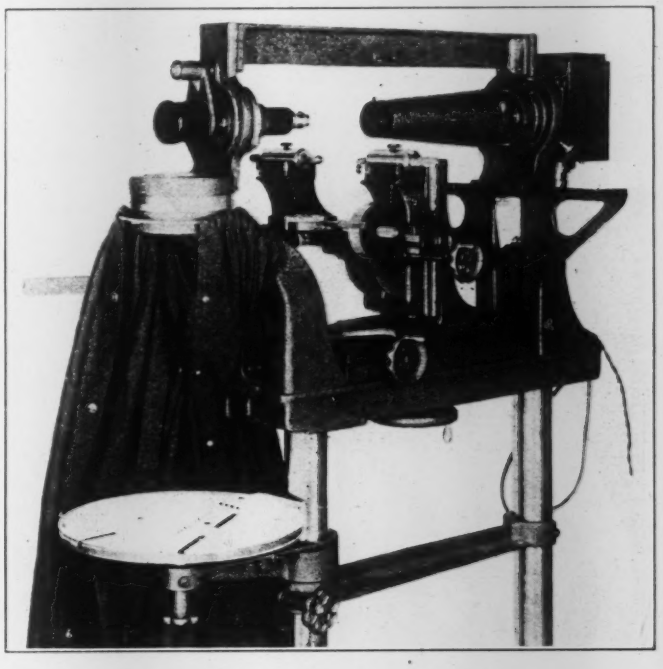

FIG. 8 CONTOUR-MEASURING PROJECTOR

(Of American manufacture. Affords precise measurements of flank angle, angle of "lean," lead, pitch, etc. of screw threads. May also be used as a comparator in mass production. For measurement and study of gear-tooth contour and action in mesh. For measurement of hobs, dies, punches, rolls, stampings, etc. Photographs may be made for record or study at leisure.)

or its plane contains the axis of the cylinder. The cutting tool is now set to the templet with the help of the magnifier. The fine line of light between the tool edge and the templet makes an excellent criterion by which to judge the angle and position of the tool.

With either of the above devices the angle of the cutting tool and its angular position may be judged within a few minutes of arc, and with the latter it may be set at center height in addition, at the expenses of a few moments, for both inside and outside threads.

Fig. 4 is from a photograph of a tool-makers' microscope of American design. The microscope itself is of the erecting type, giving images and motions similar in aspect to that of the object

viewed. The eyepiece shown is provided with a simple cross-wire. The stage is of the cross-slide type, actuated by micrometers reading to 0.0001 in. Illumination is provided by a mirror train, completely protected by the base. The illustration shows a drill-jig plate in position for the measurement of the location of the holes.

Fig. 5 shows an accessory stage adapted to the measurement of screw-thread gages and screws. The screw is clamped in a V-block which may be rotated about a horizontal axis through an angle equal to the helix angle. It is so constructed that this tilt may be instantly changed to correspond with the helix at opposite sides of the screw. Fig. 6 shows a goniometer eyepiece for use with this stage. This eyepiece contains two cross-lines. The angle between these cross-lines may be measurably varied, and the bisector of the cross may be measurably adjusted to the axis of the screw. These accessories make it possible to measure the flank angle and the angle of "lean" to within ten minutes of arc, and the external diameter, the sharp external diameter, the root diameter, the sharp root diameter, the lead, and the pitch to 0.0001 in. These constants are not determined by comparison with a standard, but are measured directly, no comparison standard being required. Further, a critical inspection of the smoothness, finish, and contour of the surface is possible. These measurements can be made in a short time, and the procedure is simple enough to be easily accomplished by the ordinary mechanic.

Previous to the introduction of optical screw-thread measuring apparatus within a very few years, the engineer had no dependable means for measuring all of the values necessary to the complete specification of the screw. The search for means to measure each of the dimensions independently of the others led to the design of optical projection apparatus capable of it. There are two kinds: first, comparators, by means of which the projected image of a thread is compared with a standard outline arrived at by any suitable means; and second, direct measuring apparatus, which does not require any such standard of comparison.

Fig. 7 shows a line drawing of a screw-thread comparator of American manufacture. The screw thread is placed in a suitable holder. It is illuminated and its contour projected on a trans-

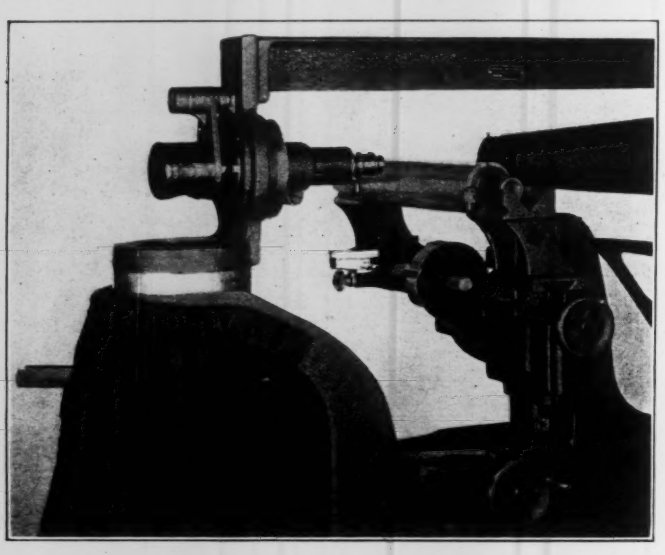

FIG. 9 CONTOUR-MEASURING PROJECTOR WITH SCREW IN PLACE
(Screw in position for projection and measurement either on horizontal screen which is part of instrument, a photographic plate, or a wall screen.)

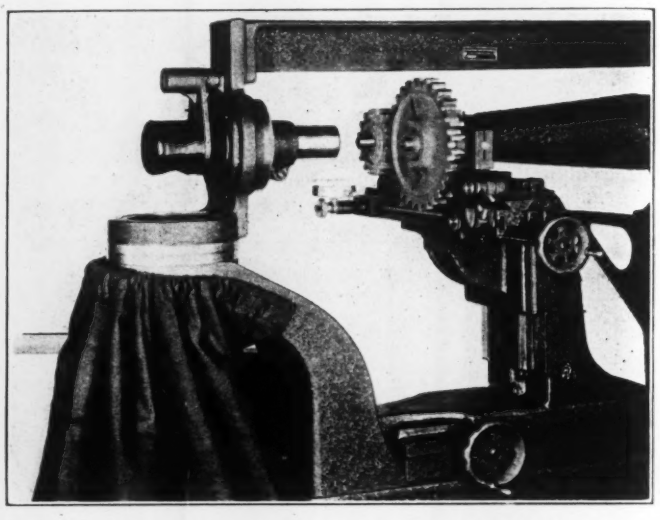

FIG. 10 CONTOUR-MEASURING PROJECTOR WITH A PAIR OF SPUR GEARS IN PLACE
(Gears in position for projection for measurement and study.)

FIG. 11 CALIPER WITH TRANSPARENT SCALE
(The scale is completely enclosed and protected against abrasion, shock, and wear. A single-line index and magnifier are used for reading scale, the least interval being 1/2 in. This least interval may be made any desirable quantity within limits.)

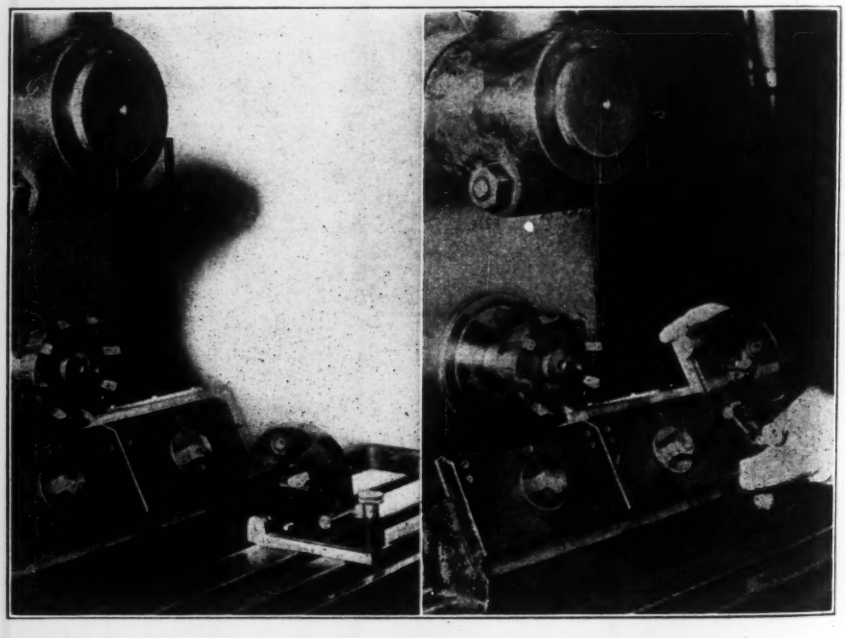

FIG. 12 TWO VIEWS OF A LEVEL PROTRACTOR
(All measurements referred to horizontal. Scale completely enclosed. Least interval 10 min. for single-line index, 1 min. for vernier reading. Angular positions of widely remote parts of machines may be accurately checked.)

lucent screen by means of a microscope and mirror system, placed conveniently with respect to the operator. This screen has on it a limiting contour drawn in such a manner as to establish "go" and "no go" limits for screws of the pitch in question. These limit charts are removable, and one of any characteristics may be used. The whole instrument is very handy and lends itself admirably to the critical inspection of screws in mass production.

Fig. 8 shows another machine made by another American manufacturer which is designed to afford facility for making measurements, not comparative but direct, of flank angle, lean, pitch, lead, etc., to within the same limits as the tool-makers' microscope previously described, in addition to serving as a complete comparison instrument in mass-production work. It further serves for the measurement and comparison of gear teeth as regards their dimensions, forms, and mesh performance, and for the inspection and measurement of rolls, dies, punchings, levers, hobs, and many other parts.

Fig. 9 shows the instrument with a screw in place and ready for inspection and measurement.

Fig. 10 shows a pair of spur gears in place for examination.

Special equipment is provided for the examination of spiral gears, and for objects requiring incident illumination instead of transillumination.

Table 1 is interesting in that it shows the kind and magnitude

TABLE 1 MEASUREMENTS ON SCREWS

(Measurements on five screws, all of which passed inspection for fit to thread ring gage by an experienced inspector. Note effect of "lean" and flank-angle errors on 3-wire measurements.)

	Screws				
	A	B	C	D	E
External diameter (micrometer)	0.5004	0.5000	0.5000	0.4995	0.5006
Pitch diameter (3-wire)	0.4505	0.4498	0.4493	0.4468	0.4440
Pitch diameter (microscope)	0.4505	0.4474	0.4468	0.4453	0.4420
Flank angle (microscope)	60°	58° 47′	58° 47′	60°	59° 5′
"Lean" (microscope)	0°	0°	0°	2° 10′	0° 30′

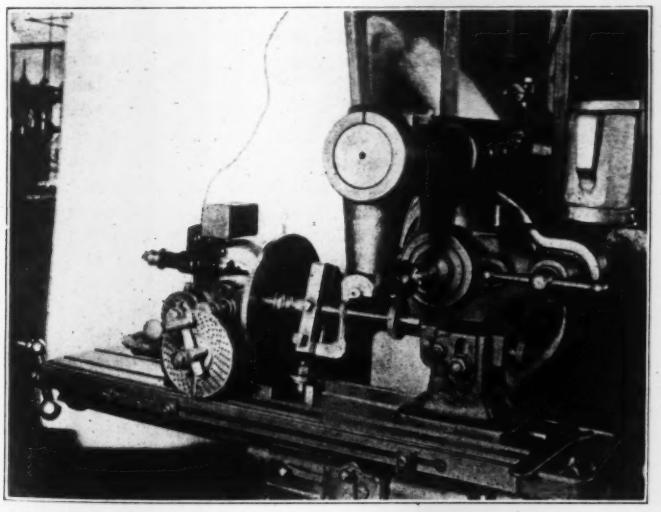

FIG. 13 DEVICES FOR SETTING MILLING-MACHINE HEAD

(Attachable head and reading microscope. Least reading, 1 min. of arc. No mechanism between scale and spindle. Indexing may be checked after clamping.)

of errors disclosed by optical methods of thread measurement. The value of the method for precision work, especially in the making of gages and masters, is apparent. The data were taken on four screws, all of which were accepted and passed for fit to a ring screw gage, by an unbiased and experienced inspector. They were then measured by the three-wire system and by the optical method. The differences between the screws as disclosed by the optical method are shown in the table.

OPTICAL SCALE READING

The angle-measuring tools used by the tool maker are usually provided with scales and verniers divided on steel. The verniers are hard to read for the average mechanic, the lines are coarse and conducive to inaccuracy, and the scales are susceptible to damage by abrasion and corrosion. Advantage of magnification has been taken recently in the use of a glass scale, divided to fractions of a degree, and a single-line index by means of which angles may be accurately estimated to 5 minutes of arc. For linear measurements the scale is divided to read fractions of an inch, either by means of a single line or an eyepiece micrometer. The scale is completely enclosed and protected against abrasion, and the elimination of the vernier reduces the setting and reading to the lowest terms of simplicity.

Fig. 11 shows a caliper made with such a scale divided to read $^1/_{32}$ in. The application of a scale to this handy tool is in itself a step forward, and its many uses are obvious. Its use often eliminates the need for a transfer caliper.

Fig. 12 shows two views of a level protractor, the uses of which

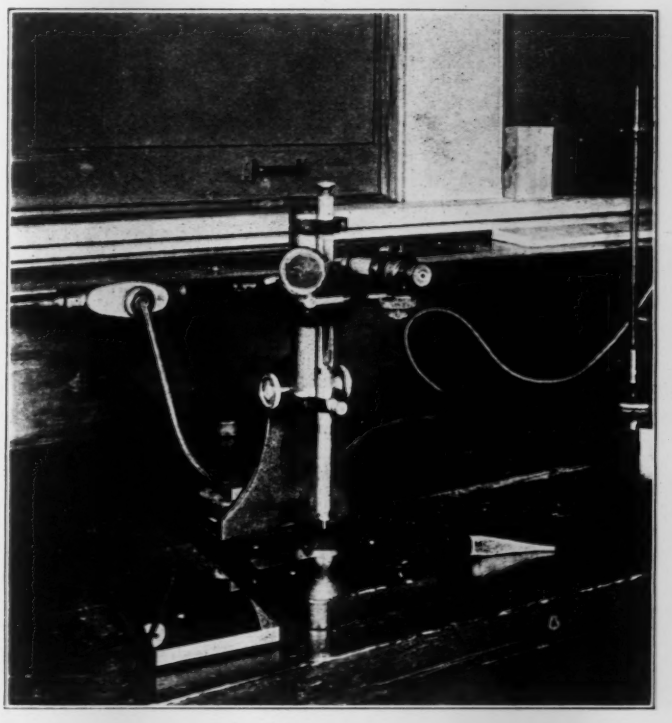

FIG. 14 THICKNESS GAGE

(Of European manufacture. Least reading, 1 micron or $^1/_{25,000}$ in. Scale attached directly to plunger. Protected by glass cover and read with micrometer eyepiece.)

in setting up work on machines are manifold. It is also useful for checking the squareness of the beds on machines and the angular relations of machine parts not easily measured otherwise. The combination of level and scale permits the reference of all angles to the horizontal. The protractor is accurate to half the least reading of the scale, 5 minutes of arc. For very precise work a short vernier is provided reading to a single minute of arc.

Fig. 13 is a photograph of a milling-machine indexing head, fitted with an angular scale and a reading microscope. This scale is divided in an accurate dividing machine and can be depended upon to 20 seconds of arc. The whole equipment is adapted to give angular readings to within one minute of arc. For precise indexing this equipment, which may be made attachable at will, is excellent because the scale is attached directly to the spindle, without any intervening mechanism. The actual rotation of the spindle is measured and not the rotation of some member geared to it. The setting may be checked after clamping the spindle, directly before and after cutting to assure precision.

Fig. 14 shows a precision thickness gage of European manufacture. The vertical plunger carries a scale graduated to 0.1 mm. under glass, where it is protected against damage. By means of the microscope with micrometer eyepiece thicknesses may be read to 0.001 mm. or 0.00005 in. with certainty. The rising and falling of the scale plunger is controlled by means of a cord wrapped around a cylinder rotated by handy buttons. The greatest advantage of this instrument lies in the fact that its basis of measurement, the scale, is not subject to wear or damage, and measurements may be thoroughly relied upon throughout the life of the instrument.

Fig. 15 shows a thickness gage of American manufacture based on the same fundamental principles as that shown in Fig. 14. The

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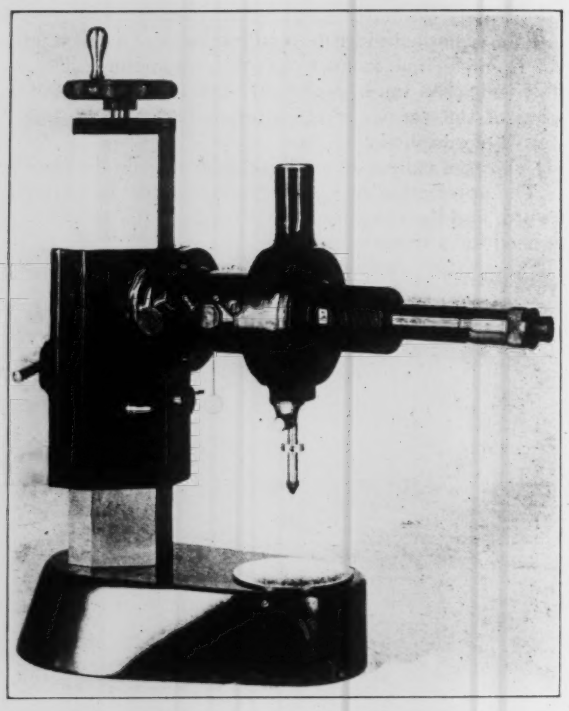

FIG. 15 THICKNESS GAGE

(Of American manufacture. Scales are transparent. Transparent scale is used for a micrometer as well as for a plunger scale, thereby relieving instrument of dependence on gears, screws, levers, or other mechanical elements.)

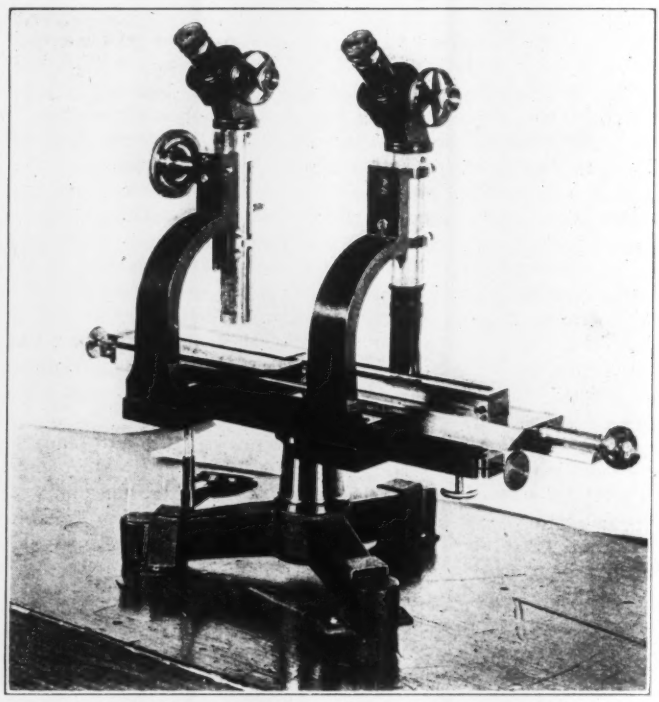

FIG. 16 LENGTH COMPARATOR

(Of European manufacture. Two micrometer microscopes, one for setting on object and the other for reading scale to 1 micron or $^1/_{25,000}$ in. Scale protected by cover glass. No error due to pressure difference possible because there is no pressure.)

instrument is ruggedly built, and is equipped with operating devices that make it appropriate for shop use. The micrometer is in itself a further extension of the glass-scale principle, with its advantage of durability. There are no screws, levers, or other parts upon which the precision of the instrument depends. Its precision will be constant throughout the life of the scales. This must be indefinite, because all moving parts are completely enclosed for protection.

Fig. 16 shows a length comparator. It is provided with two microscopes. One of these is for reading the scale attached to the object-carrying slide, and the other is for the purpose of setting on the object. Thus the measurement is relieved of errors due to differences in contact pressure. This instrument is capable of measuring to an accuracy of one micron or $^1/_{25,000}$ in. with certainty, and this accuracy may be expected throughout the life of the instrument.

CONCLUSION

The making of machinery has passed through various stages of development. The earliest one was marked by the use of an ordinary wooden scale together with whatever other devices were at hand or could be supplied by the ingenuity of the mechanic. This stage was followed by the use of the steel scale and the caliper. Next came the use of micrometers and specially designed mechanical measuring tools. These tools have served very well until comparatively recently. The engineer is now forced to make many measurements for which this equipment is inadequate, in respect of accuracy, adaptability, and facility. Optical apparatus opens an avenue for the accomplishment of many of these measurements and for the more expeditious accomplishment of measurements now possible to the engineer. Some instruments have been described indicating what has been done up to the present. The success of such specially designed optical tools together with the obvious capabilities and possibilities for further uses of optical apparatus indicate that much remains to be done before these possibilities are exhausted.

The Supply of Industrial Power

By WILLIAM HARRISON LARKIN, JR.,[1] PASSAIC, N. J.

This paper is intended as a message to power users and factory managers, with the thought that it may aid them in differentiating between good power-plant operation and inefficient work, help them to a better understanding of their engine and boiler rooms, assist them in getting maximum efficiency out of present plants, and indicate the course to be taken when it comes to enlargement or replacement. To assist in determining the efficiency of plant operation, tables of actual operating data and costs from industrial power plants scattered about the country are shown, including plants which use purchased electric power. Paragraphs are included on the subject of economy of operation which indicate the desirability of power-plant supervision from the manager's office—the need of instruments, so that what is taking place in the most expensive non-productive department of the factory can be known. The necessity and value of feedwater treatment are pointed out, and a warning is given of the danger of falling into the hands of boiler-compound men. Some space is given to the matter of adequate continuous power records, and such a complete record is shown and commented upon. The importance of accuracy in making up power costs is discussed, particularly in connection with the matter of maintaining meters and similar devices. The advisability of coal and fuel testing is suggested. Some aids to economy are given in the nature of charts showing continuous efficiency and evaporation curves and the curve of relationship between efficiency and operating cost. An attempt is made to outline methods by which the quality and relative cost of the supply of industrial power may be estimated.

THIS paper is a study of what is often the largest item of factory expense, that is, the power plant. It is an attempt to let light into a corner shrouded with mystery for the office man and something often considered too technical a matter for much factory supervision and control. The examination of an industrial power plant should be based on four principles, namely, the question of its suitability, the possibility of future enlargement, its proper and economical operation, and finally, the availability of correct and continuous operating data and cost figures.

If the power requirements are small, the modern internal-combustion engine or a motor driven by central-station electric power may best meet the needs of the factory. Should the heating load be considerable, the steam engine with available exhaust would probably be more economical. When there is a large demand for process steam the modern high-back-pressure turbine or perhaps the bleeder-type turbine may prove best.

The question of suitable boilers and fuel is important. Hand-fired boilers operated at low ratings are usually inefficient and can be replaced to advantage by modern water-tube boilers operated at higher ratings and better efficiencies.

Coal purchased may be cheap, but if transportation is high it often proves to be comparatively expensive. In New England, for instance, the higher-priced and better grades of coal usually generate more steam per dollar than the cheaper grades. A comparatively low-priced high-volatile coal must be burned with much excess air, resulting in inefficient operation, in order that the smoke may not exceed limits allowed by law. The use of fuel oil and pulverized coal may have advantages. Whether or not either can be justified is a matter requiring careful study from the investment and operating standpoints.

A well-considered power-plant program must contemplate enlargement. Many industrial plants are poorly laid out, are piecemeal growths of many years where expediency for the moment was considered above all else. Such plants are often enough found with small boilers of different types, uneconomical steam engines scattered about, sometimes several complete power plants located in different corners of the factory yard with long steam mains and transmission. Consolidation and electrification in such cases will usually pay satisfactory returns on investment and substitute an economical plan for future operation and enlargement for a haphazard method of generating power.

The day of the old-fashioned engineer who knew very little of theory and cared less is long past. The cost of fuel is too high for indifferent or careless operation, and inefficiency has too great an effect on the cost of production to pass unnoticed.

No factory can afford to neglect careful supervision of labor, raw material, and expense, of which the power house often makes up the largest item. No power plant can be considered as operating economically unless attention is paid to the matter of proper combustion, excess air, stack temperatures, feedwater treatment, feedwater heating, return of hot condensate, boiler-cleaning program, best point of rating, suitable refractories and furnace volume, economical transmission, proper lubrication, necessary operating records, and all the modern aids to continuous operation at the highest possible point of efficiency.

No power department is well operated which does not produce regular and understandable data on power operating efficiencies and costs that can be compared from month to month with records of similar plants. Operators should be made to realize that 60 per cent boiler efficiency is poor, that costs per thousand pounds of steam generated, per engine horsepower, or per kilowatt-hour will be carefully checked by the hand that controls the salary list.

Operating Costs

Figures are shown in Tables 1 and 2 which give comparative unit costs carefully compiled for the year 1924 from several factory power plants located in different parts of the country. The selection was made to show plants of varied character and sizes. These tabulations indicate a cost of steam averaging around 38 cents per 1000 lb. for plants in the Middle West, 49 cents per 1000 lb. for plants in the New York district, and 57 cents per 1000 lb. in New England.

Engine power per horsepower-hour averages 2.19 cents for the Middle-Western group, 4.42 cents for the New York group, and 5.54 cents for the New England group. Generated electric power per kilowatt-hour averages 0.693 cent for the Middle-Western group, 2.81 cents for the New York group, and 2.78 cents for the New England group. Few plants have water power and the cost runs from 0.66 cent per horsepower in one case to 2.255 cents in the other. Public-service electric power per kilowatt-hour averages 1.86 cents in the Western group, 2.73 cents in the New York group, and 2.61 in the New England group. Assuming that all the motive power is purchased from the public-service companies and using the demand and energy rates which prevail in the districts mentioned, the cost per kilowatt-hour would be approximately 1.612 cents in the Western district, 3.023 cents in the New York district, and 2.695 cents in the New England district.

In all these cases the public-service unit costs are made up of the cost per kilowatt-hour at the bus plus an operating charge which averages around 33 per cent. This operating expense covers the charges on the investment, operators' wages, and repairs and maintenance of substation, feeders, and motors.

Table 3 shows the method used in arriving at the comparative figures used in the tabulations. The object in presenting these figures is to give those who have the problem of power costs under consideration a picture of what is being done in plants of varying capacity and character, and to present a method by which definite conclusions may be arrived at.

Economy of Operation

Careful supervision over productive labor, the goods turned out by the factory, and the general matter of expense is usual and understood. Unfortunately the matter of power-plant supervision is often considered too technical for control except in a very general way. The operating engineer receives very little attention in many industries, and indifference, poor operating efficiency, and high costs often result. Many smaller power plants run along in ways that were obsolete years since, and many factories with fine buildings

[1] Power Engineer, General Division, United States Rubber Company. Mem. A.S.M.E.

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TABLE 1 INDUSTRIAL POWER DATA

Group		Location	Average No. Employees Prod. & Non Prod.	Kind Of Power Used	Annual Total Equivalent K.W. Hours	Total Cost Power (Boilers Not Included)	Cost Per Equivalent K.W. Hour	Estimated Cost All On Pub. Service Power
Central Group								
	1.	Indiana	368	Engine 20% Wat. Pow. 14% Gen. Electric 66%	11'892'211	119'512.42	.01005	.01686
	2.	Ohio	1203	Engine 5% Pub. Serv. Elect. 95%	7'013'395	143'724.91	.0205	——
	3.	Illinois	381	Engine 19% Pub. Serv. Elect. 81%	2'876'500	75'522.96	.02626	——
	4.	Michigan	3985	Engine 6% Gen. Elect. 66% Pub. Serv. Elect. 28%	33'530'767	380'524.72	.01136	.0145
	5.	Indiana	1032	Engine 23% Pub. Serv. Elect. 77%	5'155'372	94'096.23	.01825	.017
	6.	Maryland	——	Gen. Electric 100%	11'491'000	38'725.00	——	——
	7.	Penn.	——	Gen. Electric 100%	8'518'000	44'464.00	——	——
	8.	Louisiana	——	Gen. Electric 100%	9'598'000	40'983.00	——	——
N. Y. City Group								
	9.	New Jersey	1368	Engine 3% Pub. Serv. Elect. 97%	8'019'000	227'695.20	.0284	——
	10.	Conn.	328	Engine 50% Gen. Elect. 3% Pub. Serv. Elect. 47%	10'143'078	185'217.65	.0183	.01785
	11.	New Jersey	343	Engine 31% Gen. Elect. 5% Pub. Serv. Elect. 64%	420'447	23'339.03	.0557	.0426
New England Group								
	12.	Mass.	1136	Engine 3% Pub. Serv. Elect. 97%	5'291'000	134'292.86	.0254	——
	13.	Conn.	300	Engine 41% Wat. Pow. 25% Gen. Elect 13% Pub. Serv. 21%	1'697'466	54'488.67	.0321	.02668
	14.	R. I.	613	Engine 1% Pub. Serv. Elect. 99%	3'847'598	111'459.85	.029	——
	15.	R. I.	919	Engine 24% Pub. Serv. Elect. 76%	3'842'000	102'790.23	.0267	.0324
	16.	Mass.	3150	Engine 80% Pub. Serv. Elect. 20%	2'601'023	——	——	.0275
	17.	Mass.	285	Pub. Serv. Elect. 100%	49'338	2'077.00	.0421	.0421
	18.	Mass.	1050	Gen. Elect. 86% Pub. Serv. Elect. 14%	1'294'778	21'566.75	.0167	.022
	19.	Conn.	1543	Pub. Serv. Elect. 100%	11'663'000	208'809.17	.01792	——
	20.	R. I.	159	Engine 54% Gen. Elect. 6% Pub. Serv. Elect. 40%	1'071'778	31'279.92	.0182	.039
	21.	Mass.	3700	Pub. Serv. Elect. 100%	31'206'000	396'075.00	——	.0125
	22.	Mass.	——	Gen. Electric 100%	6'631'000	37'497.00	——	.01342

NOTE:

NEXT TO LAST COLUMN – COST PER EQUIVALENT K.W. HOUR MEANS COST OF ALL CLASSES OF POWER AS AT PRESENT DIVIDED BY THE EQUIVALENT K.W. HOURS.

TABLE 2 INDUSTRIAL POWER DATA

Group		Location	Average No. Employees Prod. & Non Prod.	Kind Of Power Used	Unit Costs For Power 1924				
					Steam Per 1000 Pounds	Engine Power H.P. Hour	Generated Power K.W. Hour	Water Power H.P. Hour	Public* Service K.W. Hour
Central Group									
	1.	Indiana	368	Engine 20% Wat. Pow. 14% Gen. Electric 66%	.316	.0115	.00885	.0066	——
	2.	Ohio	1203	Engine 5% Pub. Serv. Elect. 95%	.327	.0417	——	——	.01922
	3.	Illinois	381	Engine 19% Pub. Serv. Elect. 81%	.4929	.03	——	——	.0229
	4.	Michigan	3985	Engine 6% Gen. Elect. 66% Pub. Serv. Elect. 28%	.378	.0111	.00994	——	.01453
	5.	Indiana	1032	Engine 23% Pub. Serv. Elect. 77%	.3843	.0153	——	——	.0177
	6.	Maryland	——	Gen. Electric 100%	——	——	.00437	——	——
	7.	Penn.	——	Gen. Electric 100%	——	——	.00622	——	——
	8.	Louisiana	——	Gen. Electric 100%	——	——	.00527	——	——
N. Y. City Group									
	9.	New Jersey	1368	Engine 3% Pub. Serv. Elect. 97%	.462	.0669	——	——	.0253
	10.	Conn.	328	Engine 50% Gen. Elect. 3% Pub. Serv. Elect. 47%	.398	.0143	.0392	——	.0162
	11.	New Jersey	343	Engine 31% Gen. Elect. 5% Pub. Serv. Elect. 64%	.609	.0515	.01695	——	.0403
New England Group									
	12.	Mass.	1136	Engine 3% Pub. Serv. Elect. 97%	.5940	.0644	——	——	.01755
	13.	Conn.	300	Engine 41% Wat. Pow. 25% Gen. Elect 13% Pub. Serv. 21%	.5550	.0288	.025	.02255	.0272
	14.	R. I.	613	Engine 1% Pub. Serv. Elect. 99%	.5700	.126	——	——	.0264
	15.	R. I.	919	Engine 24% Pub. Serv. Elect. 76%	.5260	.0332	——	——	.0214
	16.	Mass.	3150	Engine 80% Pub. Serv. Elect. 20%	.685	——	.0287	——	.0284
	17.	Mass.	285	Pub. Serv. Elect. 100%	——	——	——	——	.0421
	18.	Mass.	1050	Gen. Elect. 86% Pub. Serv. Elect. 14%	.5707	——	.0368	——	.0331
	19.	Conn.	1543	Pub. Serv. Elect. 100%	.5741	——	——	——	.01792
	20.	R. I.	159	Engine 54% Gen. Elect. 6% Pub. Serv. Elect. 40%	.5420	.0246	.042	——	.0344
	21.	Mass.	3700	Pub. Serv. Elect. 100%	.532	——	——	——	.0125
	22.	Mass.	——	Gen. Electric 100%	——	——	.0067	——	——

* UNIT COST OF PUBLIC SERVICE POWER INCLUDES PUBLIC SERVICE BILLS, PLANT OPERATION, REPAIRS AND MAINTENANCE OF APPARATUS, FEED LINES, ETC., DEPRECIATION AND INSURANCE CHARGES. USUALLY EQUALS PUBLIC SERVICE CHARGE PLUS 20 TO 30% (AVERAGE).

and excellent machinery shelter power plants with scaled-up boilers, tubes covered with soot, leaky furnaces, broken baffles, engines with valve gear out of adjustment, hot returns thrown overboard, and similar losses which run on because of the lack of a little understanding and supervision from the office. No modern factory can afford to be without the instruments which indicate what is going on in the power plant and which will turn their messages of character of daily operation into the office continuously. There will be financial loss if the stack temperatures, excess air, CO_2, draft, proper rating, and coal and water consumed are allowed to run along unchecked and unnoticed.

If no attention is paid to the matter of feedwater treatment and boiler scale, it will be found expensive. Many a boiler steams along with an eighth or a quarter of an inch of scale on its tubes, using perhaps 25 per cent more coal than is necessary. Many an engineer follows the obsolete scheme of dosing the boilers with a compound sold him by some irresponsible salesman at high cost and little result. The modern engineer uses the up-to-date scheme of analyzing his raw water frequently, testing his steaming boilers for alkalinity and chloride, and works under the supervision of some reputable feedwater engineer or specialist.

If firebrick in furnace walls are neglected, the result may be an entire new setting at large expense.

TABLE 3 SHOWING METHOD USED IN COMPARING PRIVATE OPERATION WITH 100 PER CENT PURCHASED ELECTRIC POWER

Plant No. 11, N. J., 1924

	Total Cost	Unit Cost	Total Power Used	Equiv. K.W. Hours
Engine Power	8812.98	.0515	171'280	128'500
Generated Electric	3'680.52	.01695	21'669	21'669
Purch. Elect. Power	10'899.53	.0403	270'278	270'278
Water Power	——	——	——	——
Total	23'393.03	——	——	420'447
Avg. Cost Equiv. K.W. Hr.	(As Operating at Present)			.0557

Plant No. 15, R. I.

	Total Cost	Unit Cost	Total Power Used	Equiv. K.W. Hours
Engine Power	40'010.23	.0332	1'205'384	904'000
Generated Electric	——	——	——	——
Purch. Elect. Power	62'780.00	.0214	2'938'000	2'938'000
Water Power	——	——	——	——
Total	102'790.00	——	——	3'842'000
Avg. Cost Equiv. K.W. Hrs.	(As Operating at Present)			.0267

Below Are Estimates Based On The Supposition That All The Power Is Taken From The Public Service Electric Company At Wholesale Rates.

Plant #11 N. J. Comparison, All On Public Service.

Total Equiv. K.W. Hours		420'447
Average Per Month K.W. Hours		35'037
Average Load @ 172 Hours Per Month, K.W.		203
Max. Demand Estim. 150% ± (300 K.W. = 400 H.P.)		400
Demand Charges		
200 H.P. @ 1.50	300.00	
200 H.P. @ 1.35	270.00	
Energy Charge		570.00
3800 K.W. Hrs. @ .03	90.00	
7000 " " @ .02	140.00	
25037 " " @ .01	250.37	
		480.37
Coal Clause 35'037 K.W. Hrs. @ .00085		29.78
Total Cost For One Month (Public Service Bill)		#1080.15
Cost Per K.W. Hr. (Public Service Bill)		.0308
Operating Charge 38½%		.0118
Total Cost To Plant Per K.W. Hour		.0426

Plant #15 R.I. Comparison, All On Public Service.

Total Equiv. K.W. Hours		3'842'000
Average Per Month K.W. Hours		320'166
Average Load @ 172 Hours Per Month, K.W.		1'860
Max. Demand Estim. 150% ± K.W.		2'800
Demand Charges		
200 K.W. @ 2.00 =	400.00	
100 " @ 1.85 =	185.00	
200 " @ 1.75 =	350.00	
250 " @ 1.65 =	412.50	
250 " @ 1.55 =	387.50	
500 " @ 1.45 =	725.00	
500 " @ 1.35 =	675.00	
800 " @ 1.25 =	1000.00	
Energy Charge		4135.00
320'166 K.W. Hours @ .012		3841.99
Total Cost For One Month (Public Service Bill)		7976.99
Cost Per K.W. Hr. (Public Service Bill)		.0249
Operating Charge 30%		.0075
Total Cost To Plant Per K.W. Hour		.0324

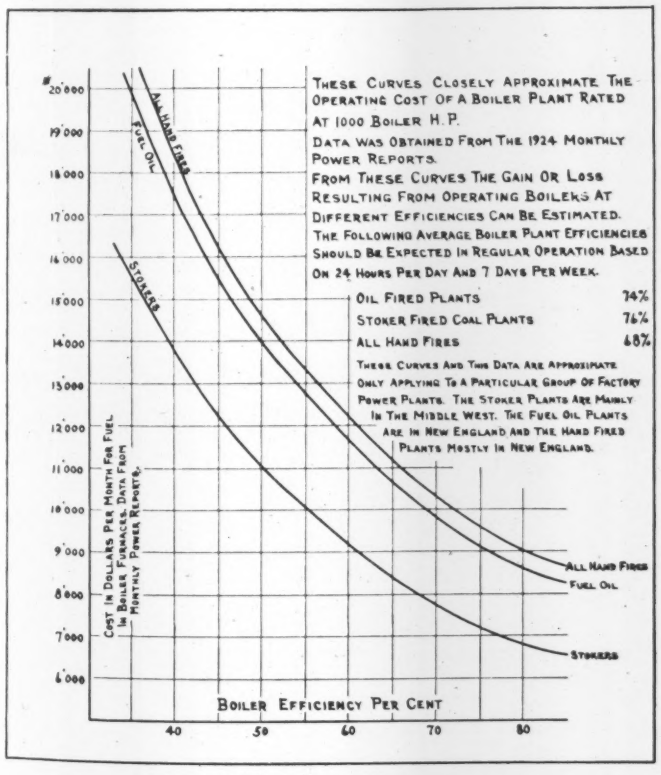
These Curves Closely Approximate The Operating Cost Of A Boiler Plant Rated At 1000 Boiler H.P.
Data Was Obtained From The 1924 Monthly Power Reports.
From These Curves The Gain Or Loss Resulting From Operating Boilers At Different Efficiencies Can Be Estimated.
The Following Average Boiler Plant Efficiencies Should Be Expected In Regular Operation Based On 24 Hours Per Day And 7 Days Per Week.
Oil Fired Plants 74%
Stoker Fired Coal Plants 76%
All Hand Fires 68%
These Curves And This Data Are Approximate Only Applying To A Particular Group Of Factory Power Plants. The Stoker Plants Are Mainly In The Middle West. The Fuel Oil Plants Are In New England And The Hand Fired Plants Mostly In New England.
All Hand Fires
Fuel Oil
Stokers
Cost In Dollars Per Month For Fuel In Boiler Furnaces, Data From Monthly Power Reports.
Boiler Efficiency Per Cent
20 000
19 000
18 000
17 000
16 000
15 000
14 000
13 000
12 000
11 000
10 000
9 000
8 000
7 000
6 000
40
50
60
70
80

FIG. 1 BOILER EFFICIENCY AND OPERATING EXPENSE
(Estimated from Power Report data, 1924.)

An economical operating engineer will see that a regular schedule of boiler inspection and cleaning is carried out. He will arrange to have boilers off the line at stated intervals, and while he looks after the details of the work, will expect a certain amount of supervision.

The lubrication of shafting and engines and the reduction of friction load are all matters which can receive attention with profit. Most oil concerns are glad to have their engineers examine any plant, and make a detailed report as to conditions found. It is seldom that such an examination and report does not reduce unnecessary friction load, quantity of oil and grease used, eliminate unsuitable lubricants, and save money.

Turbines with eroded and broken blades, condenser air leaks, reduced speed, poor power factor, and wet or improper steam supply build up expense and coal consumption imperceptibly but rapidly. These things cannot be detected on the instant, but regular observation of power records, frequent visits to the power plant, occasional judicious questions, and an interest in the power machinery will pay large returns in the long run. A little supervision will make a good engineer better, and if the man in charge is a poor engineer, this will become evident all the sooner. Many an inefficient operating engineer who has lost interest and who is too dull to learn is being retained in his position for sentimental reasons, with every one concerned seemingly unaware that, should the company double his salary, purchase him a mahogany desk, and promote him to a high-titled sinecure in the office well removed from the power plant and where he could do no harm, there would be much money saved. Minute waste of material in process in the factory is noted and checked, but in the same factory, often enough, there will be a large waste of expensive fuel, shoveled at will by ignorant firemen into boilers unfit for service. A 1000-hp. boiler plant operated at 60 per cent efficiency, as is too often the case, which might just as well operate at 75 per cent efficiency, is costing from fifteen to twenty thousand dollars of unnecessary money annually. The chart given in Fig. 1 is helpful in this connection.

Power-Plant Operating Records

Continuous operating efficiency cannot be obtained from the power plant without some definite system of recording performance and costs. Power-plant records ordinarily vary in completeness from a much-thumbed book containing weights of coal delivered to more or less elaborate sheets on which are recorded miscellaneous

facts interesting to the operating engineer alone and which lead nowhere. Few such records are helpful in determining whether or not the plant is being operated in a reasonably efficient manner. The usual recourse in such cases is to employ an outside engineer to make an examination of the plant and a short and elaborate series of tests which determine more often what might be done under favorable circumstances rather than what actually is done from month to month. Such examinations and reports have their value, but more necessary is some regular and continuous method of reporting data that will give a definite relationship between performance and cost which can be compared from month to month, and with outside sources of information. Such a report should give the ratio between cost of power and pounds or quantity of goods produced. It should contain such units as the cost of steam per thousand pounds, evaporation per pound of fuel, engine power per horsepower hour, electric power per kilowatt-hour, etc. Enough operating data should be recorded so that the report can be checked. With a series of such reports, perhaps covering performance for several years, information will be available which will show whether the power department is forging ahead or losing ground in operating efficiency. Faced with the necessity for enlarged capacity, replacement of obsolete units, or the possibility of turning to purchased electric power, the management has operating costs and data which will help determine the course. Any association of power-plant engineers, to derive maximum benefit from their meetings, must have some comprehensive form of continuous power report which will allow comparison of performance data and costs.

MONTHLY POWER REPORT

PLANT ST. JO
LOCATED INDIANA
MONTH JANUARY 1925

SECTION 1, GENERAL DATA

PRODUCTION DATA

1. Pounds of Goods Produced 2'109'033 — Pounds Raw Material Used
2. Total Coal Burned Bitum. 4023.7 — Per Ct. 100 — Cost Mine 1.50 — Frt. 1.56 — Handling .42 — Total 3.48 — B. T. U. 11967
3. Total Coal Burned Anth. — Per Ct. — Cost Mine — Frt. — Handling — Total — B. T. U.
4. Total Oil Burned Bbls. — Specific Gravity — Cost Delivered — B. T. U.
5. Cost of all Power 25'256.62 — Same Less Charges 21'496.27
6. Cost of all Power from Accts. 19'826.48 — Total Man Hours, Productive 48 034
7. Cost of all Power per Pound Goods Produced .012 — Per Pound of Raw Material Used — Per Man Hour .526

OPERATING DATA

8. Factor of Evaporation 1.059 & 1.212 — Pounds Total Equivalent Evaporation 77'723'330
9. Temperature Outside Air, Average 19 — Average Barometer Reading
10. Temperature Flue Gas, Average 591 — Degrees Superheat, Average
11. Steam Pressure Gauge, Average 140 — Feed Temperature Fahr., Average 198
12. Temperature Water Entering Economizer, Average — Temperature Raw City Water, Average 50

FEED WATER DATA

Grains per Gal. Carbonates 10 — Sulphates 3.6 — Alkalinity Carried 42 — Chloride Carried 6.74
Chemicals Used for Treatment CAUSTIC SODA — Total Pounds for Month 4608 — 15 BAUME SOL.

SECTION 2, POWER PRODUCTION & TRANSMISSION DATA

POWER DATA	COST ITEMS	LABOR	Other than Labor	TOTAL	UNIT COSTS
BOILERS					Steam Per M Lbs. F. & A. 212°
A Average Boiler Load 3'288	1. Tot. Cost Boiler Fuel*	x x x	14025.40	14025.40	Fuel .181
B Lbs. Evap. 73'392'191	2. Other Operat'g Exp.*	4073.40	111.84	4185.24	F. R. Labor .052
C Evap. per 12000 B.T.U. 9.145	3. Boiler R'm Auxiliaries*	82.24	20.56	102.80	Supplies .002
D Evap. per Lb. Fuel 9.12	4. Total Repairs	415.56	527.39	942.95	B. R. Aux. .001
E Equiv. Evap. F. & A. 212° 9.66	5. Charges	x x x	1303.91	1303.91	Repairs .012
F Boiler Efficiency 78.3	6. Total Cost	4571.20	15 989.10	20 560.30	Charges .017
G Overall Efficiency 89.6					Total .265
MAIN ENGINES	7. Cost of Steam Used	x x x	3 348.91	3 348.91	Main Engine
H Average Load H. P. 985	8. Other Operat'g Exp.*	368.08	51.28	419.36	Power Per
I Steam per H. P. Hour 21	9. Total Repairs	408.45	127.94	536.39	I. H. P. Hour
J Total H. P. Hours 569'330	10. Charges	x x x	504.14	504.14	.00845
	11. Total Cost	776.73	4 032.27	4 809.00	
AUXILIARY ENGINES	12. Cost of Steam Used	x x x	822.24	822.24	Auxiliary Engine
K Average Load H. P. 164	13. Other Operat'g Exp.*	197.66	6.49	204.15	Power Per
L Steam per H. P. Hour 24	14. Total Repairs	21.63	25.49	47.12	H. P. Hour
M Total H. P. Hours 122'313	15. Charges	x x x	85.20	85.20	.00947
	16. Total Cost	219.29	939.42	1158.71	
ELECTRIC GENERATORS	17. Cost of Steam Used	x x x	6 236.77	6 236.77	Generated
N Average Output K. W. 1175	18. Other Operat'g Exp.*	219.45	43.62	263.07	Electric Power
O Steam per K. W. Hour 25	19. Total Repairs	172.50	78.75	251.25	Per K. W. Hour
P Total K. W. Hours 874'534	20. Charges	x x x	1122.87	1122.87	.009
	21. Total Cost	391.95	7 482.01	7 873.96	
PURCHASED ELECT. POWER	22. Tot. Pub. Service Bills*	x x x			Purchased
Q Average Load K. W.	23. Other Operat'g Exp.*				Electric Power
R Total K. W. Hours	24. Total Repairs				Per K. W. Hour
S Average Power Factor	25. Charges	x x x			
	26. Total Cost				
WATER POWER	27. Cost for Water*	x x x	340.54	340.54	Water Power
T Average Load H. P. 184	28. Other Operat'g Exp.	285.92		285.92	Per H. P. Hour
U Total H. P. Hours 110 626	29. Total Repairs	71.55	20.33	91.88	
V Hours Operated 601	30. Charges	x x x	544.23	544.23	.0114
	31. Total Cost	357.47	905.10	1262.57	

SECTION 3

Distribution and Use of all Live Steam Generated in Thousands of Pounds for Month.

(a) Total Live Steam Used for Power	37'136 M Pounds
(b) Total Live Steam Used for Auxiliaries	3'669 M Pounds
(c) Total Live Steam Used for Heating	1'467 M Pounds
(d) Total Live Steam Used for Process Work	30'501 M Pounds
(e) Total Live Steam Used for Losses	619 M Pounds
(f) Total Live Steam Used (see Section 2, Item B)	73'392 M Pounds

NOTES—* Items marked with Star form Total Charge to Power Expense as per Form
Total Repairs and Maintenance Charges on this Report should balance with Amount charged to Dept.

Signed Nathan S. Fessenden
Factory Manager

Fig. 2 Monthly Power Report

Industrial companies operating several plants have found it advantageous and necessary to inaugurate monthly power reports for the reasons mentioned. One corporation operating several plants located in various parts of the country has used such a system of reports with beneficial results for several years. Such a report is shown as Fig. 2. It will be noted that the monthly power report shown is largely financial in character, with sufficient operating data so that it may be checked and power loads compared. Part of the recorded data come from the accounting department, where certain ledger accounts are broken down in such a manner as to make the collection of the necessary figures convenient. The remainder of the data come from the chief engineer's records. The report is made up usually by the plant engineer's assistant, although in many cases the chief engineer may attend to it. The amount of labor involved in making up such a report is not excessive, as the basic facts are kept in some form or the other in practically all power plants of any size, and the time employed is well expended on the part of the operating staff. The definite knowledge of the results of his operation and the costs of power generated tend to make the operating engineer more attentive to his duties and more interested in getting the best possible results. The fact that the report is to be scrutinized by the manager and compared with previous reports or possibly reports of other plants in the same system, keeps him alert to economy of operation. He knows that his operating efficiency is usually charted, and that any falling off will be the subject of inquiry. Such a system of continuous power reports will save thousands of dollars annually, and few plants are too small for something of the kind.

The details of such a monthly power report as the one shown are readily understandable and very little explanation is required. It is intended to be complete enough to give a true picture of the operation as a whole, and is designed to be broad enough to cover plants with several classes of power apparatus and at the same time definite enough so that the unit costs, such as fire-room labor, for instance, can be compared with those of other plants. The item of overhead expense or charges is one that perhaps requires the most study, and a method is shown in Fig. 3 which is used in many industrial power plants to distribute such expense as the proper proportion of the plant engineer's salary, the chief engineer's salary, depreciation of buildings and machinery, insurance, taxes, etc. A charge for interest on investment is not shown as many accountants do not use that item in this connection.

Accuracy of Operating Data

The majority of industrial plants keep power records of some kind, but another phase of the matter is too often overlooked, namely,

INSTRUCTIONS ON MONTHLY POWER REPORT

TABLE OF POWER DEPARTMENT OVERHEAD ITEMS

		ADD THESE TO THE ITEM "OTHER OPERATING EXPENSE," LABOR ON THE POWER REPORT		ADD THESE ITEMS TO THE ITEM OF CHARGES "OTHER THAN LABOR" ON THE POWER REPORT				TOTALS
		DISTRIBUTION OF SALARIES		DEPRECIATION		INSURANCE	TAXES	
	VARIOUS KINDS OF PLANTS	M. M.	Chief Engineer	Machinery	Buildings	On Total Reproductive Value of Machy. & Bldgs.	On Tax Assessor's Valuation Machy. & Bldgs.	
1	POWER PLANT INCLUDES	Per Cent of Entire Salary		On Reproductive Value from Appraisal Book				
	(a) Boilers	Distribute 20% of M. M's. Salary according to the amount of Connected Load	Distribute all of Chief Eng'rs Salary according to the amount of Connected Load	5%	4% Wood 3% Brick 2% Concr. Buildings	At about 1-10 of One Per Cent Consult Plant Manager	Assessors Valuation Times Tax Rate Consult Plant Manager	
	(b) Engines			5%				
	(c) Generators			6%				
	(d) Purchased Electric Power			5%				
	(e) Water Power			6%				
	(f) Inert Gas			5%				
2	POWER PLANT INCLUDES							
	(a) Boilers			5%				
	(b) Engines	Same	Same	5%	Same	Same	Same	
	(c) Generators			5%				
	(d) Purchased Electric Power			5%				
	(e) Inert Gas			5%				
3	POWER PLANT INCLUDES							
	(a) Boilers			5%				
	(b) Engines	Same	Same	5%	Same	Same	Same	
	(c) Generators			5%				
	(d) Inert Gas			5%				

4 NOTE—The above gives three typical groups of power units. Arrange yours in a similar way. To get the percentages for use in distributing salaries, proceed as follows: Say all the boilers are rated at 2,000 H. P.; all the engines, hydraulic pumps, service pumps, air compressors, steam end of turbines, etc., are rated at 1,000 H. P.; all the motors, lighting circuits, etc., connected to the plant generators rated at 1,000 H. P.; all the motors, lighting transformers, rotary converters, motor generator sets, and other apparatus connected to the public service or purchased power at 1,500 H. P.; all the load on the water wheels rated at 400 H. P.; the load made by all the inert gas compressors, motors or other apparatus rated at 100 H. P. Total, 6,000 H. P. Then boilers are charged with 33⅓% of chief engineer's salary, engines 16⅔%; generator 16⅔%; purchased power 25%, water power 6⅔% and inert gas 1⅔%. Twenty per cent of master mechanic's salary distributed the same way.

FIG. 3 INSTRUCTIONS ON MONTHLY POWER REPORT

accuracy of data. If the coal scales are incorrect, and the feed-water meter reads fast, the whole basis of costs will be wrong. Venturi meters have been found reading fifty and even a hundred per cent more than they should because of scale in the throat. Fuel-oil meters are seldom exactly correct and steam-flow meters need calibration. The same is true of pyrometers, wattmeters and all classes of power-plant instruments. No boiler plant should be considered complete without weigh tanks with which the feed-water meters can be regularly and conveniently checked. Such apparatus may consist of a pair of drums with a leveling pipe between them, holding a known weight of water at a certain temperature and emptying into a conveniently placed suction tank, or it may consist of a large tank holding enough for several hours or a day's run. In such tests correction must be made for temperature, and the flow must be continuous through the meter. Check tests of this character should be made reasonably often, from once a month to perhaps a minimum of once a quarter, depending on the character of the plant. Fuel-oil meters should be tested frequently in the same manner.

Coal scales should be checked with standard weights and care taken that they are swinging freely on their knife edges with no obstructing coal dust and dirt in the way. Steam-flow meters should check reasonably well with feed meters, and the matter of blowdown should receive careful attention. Many operators attain high boiler efficiencies due to the fact that they neglect to deduct the blowdown. The matter of coal testing is important. The operator needs to know that he is getting the most suitable fuel for the large amounts of money expended. All coal looks more or less alike, and the heat units contained in it cannot be seen. It is also important that the engineer know the thermal units in the coal that he may calculate his boiler efficiency correctly. The operating engineer may be struggling with the smoke problem, unaware that he has coal of excessively high volatile content, which a test would reveal. All power-plant instruments should be checked regularly so that their records may be true ones and the engineer may have accurate information on which to base his reports, and to guide him in the operation of his plant.

AIDS TO ECONOMY

There are many schemes which are used with more or less success to keep before those interested the operating status of the power plant. Probably the simpler the scheme, the greater its success. If some sort of gage could be devised which would record boiler operating efficiency on a moving chart, it would be of great value. Perhaps the best substitute is a monthly efficiency and evaporation chart like that shown in Fig. 4. Such a chart shows the tendency, whether to improvement or to a falling off in efficiency.

Many factory offices are provided with recording flow meters and steam gages so that load changes can be followed as a matter of interest. A chart covering fluctuations of boiler efficiency and evaporation carefully plotted, covering a year or two with fair degree of accuracy, has far greater possibilities when it is considered from the standpoint of money saving. Many operators have difficulty in appreciating the effect of boiler efficiency on the use of coal and the factory expense. A curve sheet like that shown in Fig. 1 will often assist in the promotion of plant economy.

Simple, inexpensive schemes like that shown as Fig. 5 may be devised to keep the more or less non-technical management in fairly close touch with what is going on in the power plant and help it to differentiate between good work and poor results.

Be sure to bring this copy with you to the Annual Meeting

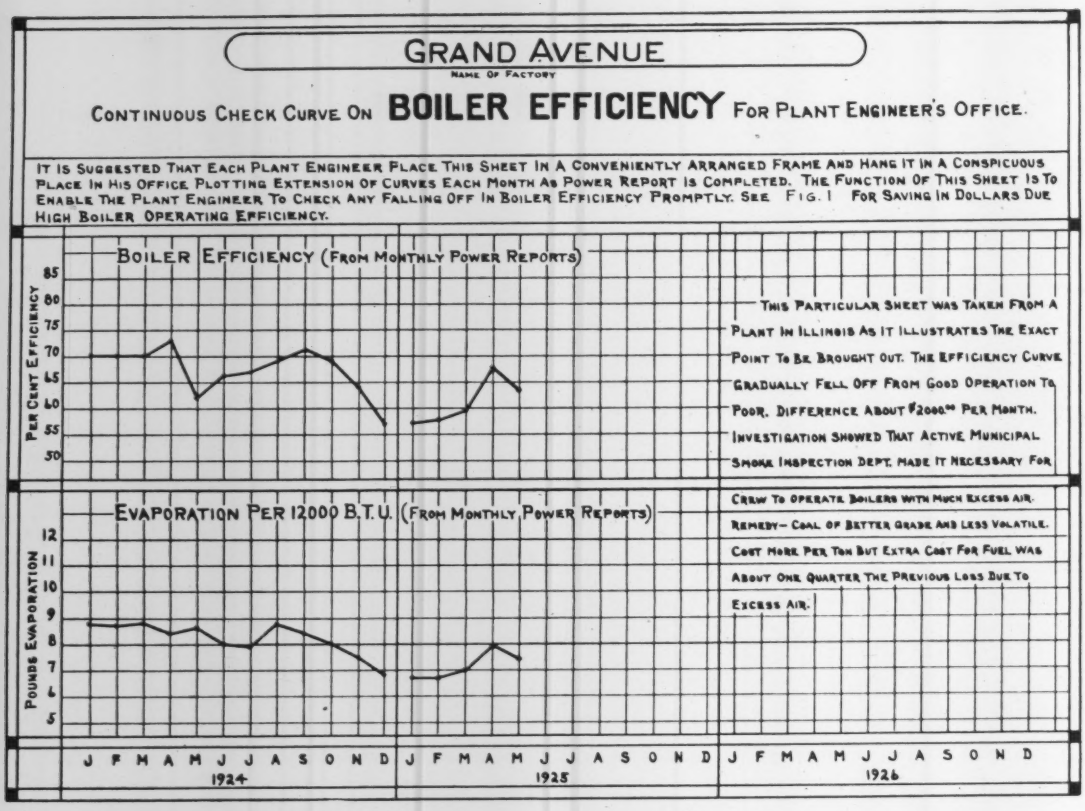
GRAND AVENUE
NAME OF FACTORY
CONTINUOUS CHECK CURVE ON BOILER EFFICIENCY FOR PLANT ENGINEER'S OFFICE.
IT IS SUGGESTED THAT EACH PLANT ENGINEER PLACE THIS SHEET IN A CONVENIENTLY ARRANGED FRAME AND HANG IT IN A CONSPICUOUS PLACE IN HIS OFFICE. PLOTTING EXTENSION OF CURVES EACH MONTH AS POWER REPORT IS COMPLETED. THE FUNCTION OF THIS SHEET IS TO ENABLE THE PLANT ENGINEER TO CHECK ANY FALLING OFF IN BOILER EFFICIENCY PROMPTLY. SEE FIG. 1 FOR SAVING IN DOLLARS DUE HIGH BOILER OPERATING EFFICIENCY.
BOILER EFFICIENCY (FROM MONTHLY POWER REPORTS)
PER CENT EFFICIENCY
85 80 75 70 65 60 55 50
THIS PARTICULAR SHEET WAS TAKEN FROM A PLANT IN ILLINOIS AS IT ILLUSTRATES THE EXACT POINT TO BE BROUGHT OUT. THE EFFICIENCY CURVE GRADUALLY FELL OFF FROM GOOD OPERATION TO POOR. DIFFERENCE ABOUT $2000.00 PER MONTH. INVESTIGATION SHOWED THAT ACTIVE MUNICIPAL SMOKE INSPECTION DEPT. MADE IT NECESSARY FOR CREW TO OPERATE BOILERS WITH MUCH EXCESS AIR. REMEDY—COAL OF BETTER GRADE AND LESS VOLATILE. COST MORE PER TON BUT EXTRA COST FOR FUEL WAS ABOUT ONE QUARTER THE PREVIOUS LOSS DUE TO EXCESS AIR.
EVAPORATION PER 12000 B.T.U. (FROM MONTHLY POWER REPORTS)
POUNDS EVAPORATION
12 11 10 9 8 7 6 5
J F M A M J J A S O N D J F M A M J J A S O N D J F M A M J J A S O N D
1924 1925 1926

Fig. 4 Continuous Check Curve on Boiler Efficiency for Plant Engineer's Office

Where it is possible, valuable comparison can be made between a number of plants in the same system, or through coöperation between a number of plants in the same neighborhood or between the plants operated by the members of some association of plant engineers, as is not unusual. Such comparisons tend to increase operating efficiency, as it is often found that what was considered good operation will not bear comparison. Such curves are shown as Figs. 6 and 7.

The Factory Power Plant

The power plants found in our mills and factories are changing in character rapidly. Twenty-five years back the fire-tube boiler and automatic engine were more or less universal. Today high-pressure water-tube boilers operated at high ratings are replacing the old horizontal-return tubular boilers. Where the engine remains in the larger plants, it has been supplemented with the low-pressure exhaust turbine which generates electricity to drive the factory machinery not in the immediate vicinity of the engine room. Frequently the engine is superseded by the high-pressure turbo-generator whose low water rate and absence of friction load and transmission losses are found profitable. In many cases the boilers only remain to generate process and heating steam, while the motive power is provided by electricity from public-service stations, coming to the factory over high-tension lines and being stepped down to safe operating limits at the factory switchboard. Each factory has an ever-present problem to determine the most economical method.

The solution of the problem of most economical method of power generation must be approached with care, as changes usually mean considerable investment. Operating costs in detail should be available over a considerable period before extensive changes are contemplated. It must be ascertained first that the present power plant is being operated to the best advantage. Poor boiler operation results in high cost of steam and high cost for motive power per unit. An attractive figure from the public-service company may in effect be the comparison of their highly efficient operation with an unnecessarily poor grade of operation in the local plant. A favorable figure from the outside power company may result, for instance, from a lack of understanding at the factory of the value of the exhaust steam used for heating and process work. There are many cases where purchased power is the most economical way out, but there is no general rule and each case must stand on its merits. When the question of plant enlargement is under consideration, the matter of public-service power, if available, should have most careful attention as then the factor of new investment is prominent. Then enlargement of prime movers will probably mean additional boiler capacity, enlarged steam mains, new coal-handling apparatus, and similar expense, which may result in an unfavorable unit cost.

Public-Service Power

In recent years numerous large electric generating units have been constructed all over the country operated by steam and water power. Long high-tension transmission lines are in evidence crossing the states for considerable distances which will be found tied into various public-service electric plants located in industrial centers. Remote localities which have a surplus of hydroelectric power send this surplus at high voltage to the cities and deliver it

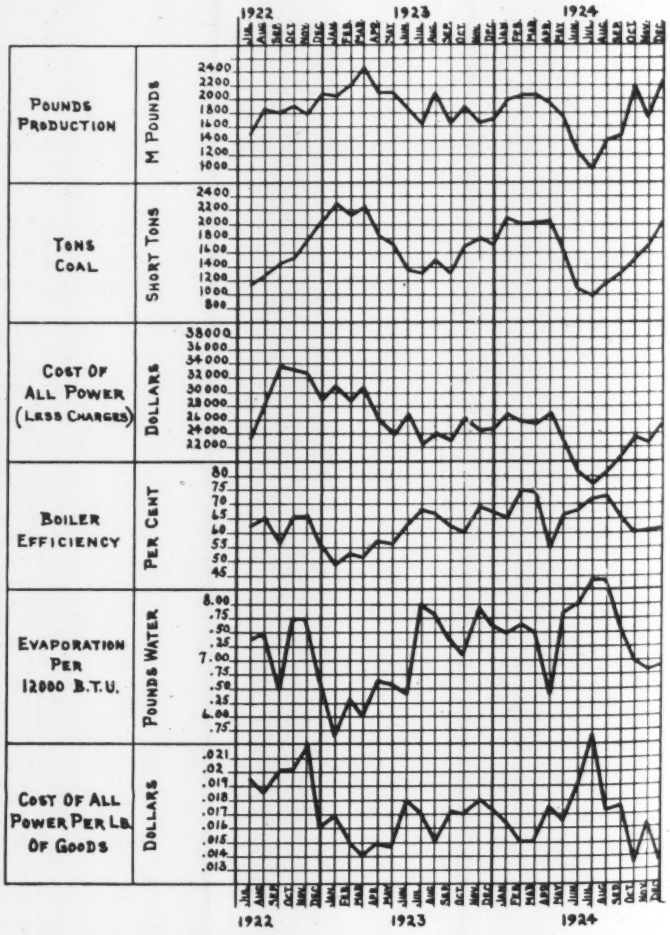

Fig. 5 Factory Manager's Power-Plant Operating Data

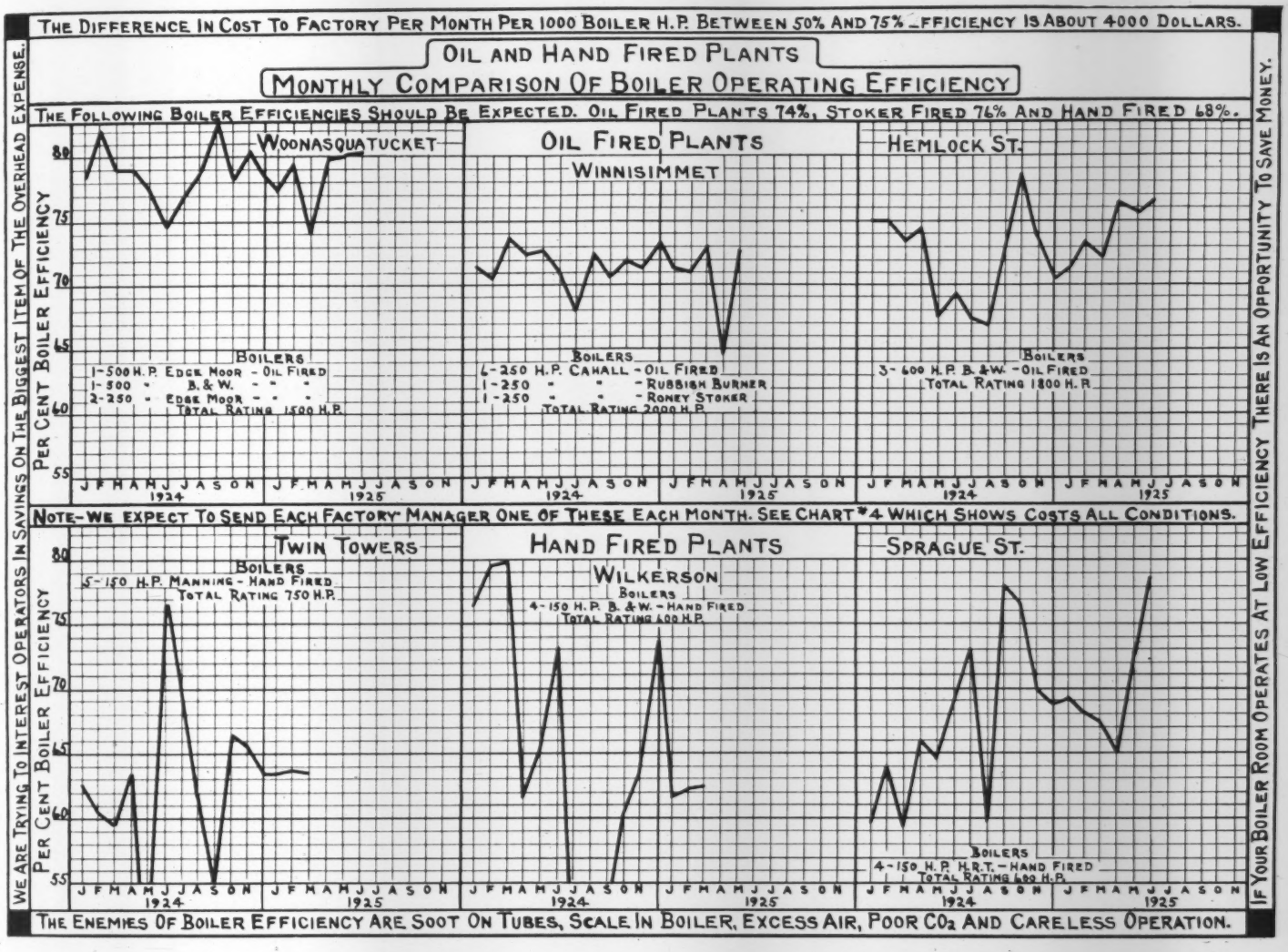
THE DIFFERENCE IN COST TO FACTORY PER MONTH PER 1000 BOILER H.P. BETWEEN 50% AND 75% EFFICIENCY IS ABOUT 4000 DOLLARS.
OIL AND HAND FIRED PLANTS
MONTHLY COMPARISON OF BOILER OPERATING EFFICIENCY
THE FOLLOWING BOILER EFFICIENCIES SHOULD BE EXPECTED. OIL FIRED PLANTS 74%, STOKER FIRED 76% AND HAND FIRED 68%.
WE ARE TRYING TO INTEREST OPERATORS IN SAVINGS ON THE BIGGEST ITEM OF THE OVERHEAD EXPENSE.
IF YOUR BOILER ROOM OPERATES AT LOW EFFICIENCY THERE IS AN OPPORTUNITY TO SAVE MONEY.
PER CENT BOILER EFFICIENCY
OIL FIRED PLANTS
WOONASQUATUCKET
BOILERS
1-500 H.P. EDGE MOOR - OIL FIRED
1-500 " B.&W. - " "
2-250 " EDGE MOOR - " "
TOTAL RATING 1500 H.P.
WINNISIMMET
BOILERS
6-250 H.P. CAHALL - OIL FIRED
1-250 " " - RUBBISH BURNER
1-250 " " - ROney STOKER
TOTAL RATING 2000 H.P.
HEMLOCK ST.
BOILERS
3-600 H.P. B.&W. - OIL FIRED
TOTAL RATING 1800 H.P.
J F M A M J J A S O N D J F M A M J J A S O N
1924 1925
NOTE-WE EXPECT TO SEND EACH FACTORY MANAGER ONE OF THESE EACH MONTH. SEE CHART #4 WHICH SHOWS COSTS ALL CONDITIONS.
HAND FIRED PLANTS
TWIN TOWERS
BOILERS
5-150 H.P. MANNING - HAND FIRED
TOTAL RATING 750 H.P.
WILKERSON
BOILERS
4-150 H.P. B.&W. - HAND FIRED
TOTAL RATING 600 H.P.
SPRAGUE ST.
BOILERS
4-150 H.P. H.R.T. - HAND FIRED
TOTAL RATING 600 H.P.
THE ENEMIES OF BOILER EFFICIENCY ARE SOOT ON TUBES, SCALE IN BOILER, EXCESS AIR, POOR CO_2 AND CARELESS OPERATION.

FIG. 6 OIL- AND HAND-FIRED PLANTS—MONTHLY COMPARISON OF BOILER OPERATING EFFICIENCY

through local public-service stations to the industrial plants of the vicinity, supplementing the power generated by the local station. This practice is so universal and so constantly increasing that the public-service stations are reaching out for an increased market for their output. The consequence has been a gradual dropping of public-service rates, until a point has been reached where any industry which generates its own power, if located where public service is available, has reason to investigate carefully the comparative costs of its operation to justify its present practice. This does not mean that no private plant can be operated profitably, but rather that many such can investigate the possibility of public service to advantage, and that few can afford to neglect careful study of how they stand with reference to this matter.

There are many general reasons why an old type of power plant often can be supplanted by outside power with profit, and the first important requisite for an investigation is the existence of reasonably accurate data which tell the story of present and past performance. It must be shown first that the plant under consideration is being operated at reasonably high efficiency, or what changes are needed to bring it to that point, and the probable cost of such changes. Only after that is done can a reasonable comparison be made with costs which would be in effect with public-service power. Many plants have several small engines distributed through the factory with long transmission lines of piping which could be supplanted profitably by electric motors. Many prime movers are old and inefficient, with high and frequently unsuspected friction loads. Many power plants are so arranged that a large number of operating engineers are required to look after the various units. In many cases the load factor is comparatively small and the friction load considerable, and the advantage of electric drive—which furnishes power at the point where it is used with less friction loss, and which can be shut down when the power is not needed—should be carefully considered. The element of safety should not be neglected. Compensation costs are carefully studied today and they are considerable in all industries. State laws are becoming more stringent continually and it is generally recognized that the electric drive, which lends itself readily to proper safeguarding, quick stopping, and ready control at all times, is an asset too important to be overlooked.

The operation of the boiler plant usually is the source of the greatest item of expense, and here again consideration of public-service power is important. Industrial boiler plants are quite generally heavily loaded, at times beyond capacity. Many are obsolete and unduly expensive to operate. Many plants are at the point where every boiler is needed on the line perhaps twenty-four hours a day, with little opportunity for proper cleaning or ordinary care. Probably the boiler plant must be enlarged in the near future, or perhaps entirely rebuilt. In such cases it will be advisable to consider outside power before entering into a program of considerable capital expense. Often privately operated generators are overloaded, and driving speeds unavoidably reduced, with increased and expensive amperage or current requirements due to the reduced driving speeds. This, too, calls for a consideration of the whole matter and of possible changes which will result in a more favorable unit cost.

The introduction of public-service power may mean the location of a transformer on a pole adjacent to the smaller plant or, in the case of a large industry, the construction of a small substation or operating unit in the factory yard. It usually results in the ultimate elimination of the old power-generating units, more room for manufacturing, simpler transmission, and the avoidance of considerable labor trouble.

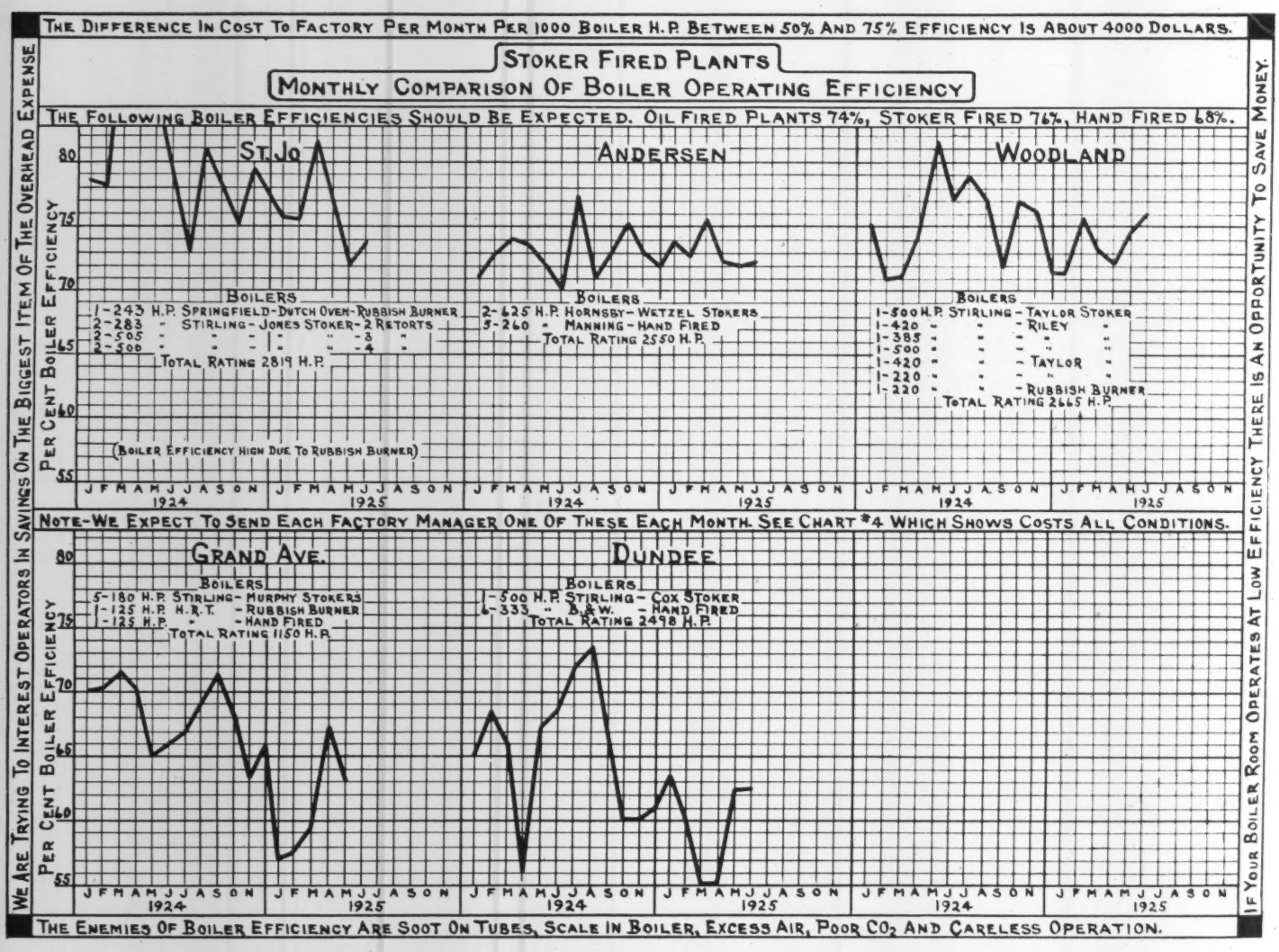
THE DIFFERENCE IN COST TO FACTORY PER MONTH PER 1000 BOILER H.P. BETWEEN 50% AND 75% EFFICIENCY IS ABOUT 4000 DOLLARS.
STOKER FIRED PLANTS
MONTHLY COMPARISON OF BOILER OPERATING EFFICIENCY
THE FOLLOWING BOILER EFFICIENCIES SHOULD BE EXPECTED. OIL FIRED PLANTS 74%, STOKER FIRED 76%, HAND FIRED 68%.
WE ARE TRYING TO INTEREST OPERATORS IN SAVINGS ON THE BIGGEST ITEM OF THE OVERHEAD EXPENSE
PER CENT BOILER EFFICIENCY
80
75
70
65
60
55
ST. JO
BOILERS
1-243 H.P. SPRINGFIELD-DUTCH OVEN-RUBBISH BURNER
2-283 " STIRLING-JONES STOKER-2 RETORTS
2-505 " " " " -3 "
2-500 " " " " -4 "
TOTAL RATING 2819 H.P.
(BOILER EFFICIENCY HIGH DUE TO RUBBISH BURNER)
ANDERSEN
BOILERS
2-625 H.P. HORNSBY-WETZEL STOKERS
5-260 " MANNING-HAND FIRED
TOTAL RATING 2550 H.P.
WOODLAND
BOILERS
1-500 H.P. STIRLING-TAYLOR STOKER
1-420 " " -RILEY "
1-385 " " " "
1-500 " " " "
1-420 " " -TAYLOR "
1-220 " " " "
1-220 " " -RUBBISH BURNER
TOTAL RATING 2665 H.P.
J F M A M J J A S O N D J F M A M J J A S O N
1924 1925
NOTE-WE EXPECT TO SEND EACH FACTORY MANAGER ONE OF THESE EACH MONTH. SEE CHART #4 WHICH SHOWS COSTS ALL CONDITIONS.
GRAND AVE.
BOILERS
5-180 H.P. STIRLING-MURPHY STOKERS
1-125 H.P. H.R.T. -RUBBISH BURNER
1-125 H.P. " -HAND FIRED
TOTAL RATING 1150 H.P.
DUNDEE
BOILERS
1-500 H.P. STIRLING-COX STOKER
6-333 " B.&W. -HAND FIRED
TOTAL RATING 2498 H.P.
IF YOUR BOILER ROOM OPERATES AT LOW EFFICIENCY THERE IS AN OPPORTUNITY TO SAVE MONEY.
THE ENEMIES OF BOILER EFFICIENCY ARE SOOT ON TUBES, SCALE IN BOILER, EXCESS AIR, POOR CO_2 AND CARELESS OPERATION.

FIG. 7 STOKER-FIRED PLANTS—MONTHLY COMPARISON OF BOILER-OPERATING EFFICIENCY

The factory substation in an average industry, say, of a thousand kilowatts capacity, often consists of a small brick or concrete building with an outside shelter for the high-tension transformers, which are usually numerous enough so that the load can be split, including one transformer for ordinary loads, another for peak conditions, and perhaps a third smaller unit for night loads, holiday, or partial shutdown periods. Mains come into the substation from underground public-service conduits to the high-tension switchboard, which is carefully enclosed, protected, and accessible ordinarily only to the public-service employees. These boards for incoming high-tension current are provided with the necessary wattmeters and recording instruments so that the quantities of power used and demand are easily read. High-tension leads run from the incoming switchboard to the transformer control board by means of which the transformer capacity can be adjusted to the needs of the factory. From this second switchboard leads run to the busbars of the distributing board, from which power is sent out to the various motor units located at various points in the factory itself. This distributing board also carries the lighting circuits, and all circuits are supplied at the board with the necessary wattmeters, control instruments, voltmeters, ammeters, power factor meters, etc., so that the whole is under the control of a single operator who is not only able to handle the situation efficiently at all times, but to keep a careful record of power consumption in the various departmental circuits. Most industries require a considerable supply of direct current to operate variable-speed motors driving elevators and variable-speed machinery. To serve this requirement the substation is generally provided with one or more rotary converters or motor-generator sets, which are also under the control of the substation operator.

The power feed lines leading from the substation to the motor units in the factory can be conveniently arranged and supported out of the way, and as a general thing are a vast improvement over the old scheme of transmission pipes, lines of shafting, and inconvenient and expensive belt transmissions which robbed the factory of light, caused excessive temperature in work rooms, were a constant source of danger to the workmen, and are better eliminated.

As a matter of interest it might be stated that a substation such as has been described has cost in several cases about twenty-four dollars per kilowatt capacity.

While it is not true that in every case an industry can turn to public-service power as a solution of its troubles, in many cases the subject can be investigated with profit. Where considerable quantities of water power are available or where large units are efficiently operated, the public-service rates possibly may be met successfully. In most cases, however, a critical comparison can be made profitable, allowing the public-service engineers ample facilities to prepare their case and making sure that the statement of the factory engineer is based on sufficient and accurate operating data.

As a matter of principle, few industries today care to go very far outside their own lines. Very few find it profitable to build their own machinery; few rubber mills, for instance, find it profitable to maintain large machine shops to manufacture the numerous molds they use in their business. Such work is largely done now by outside contract shops that make it their specialty. The rubber mill finds it profitable to confine its operating capital to the production of rubber goods; the sugar refinery, to the production of sugar; the foundry, to the production of iron castings, etc. It is becoming more and more evident that the production of power is a specialty and that such power often may be purchased with

profit from a large outside source specializing in that commodity and operating large units at a point of efficiency impossible of attainment in a comparatively small unit.

Periods of shutdown cause considerable financial loss in certain industries, particularly where large amounts of raw stock are in process at all times. Unfortunately experience of some years with public-service or central-station power has shown that there will be short periods when power is off due to electrical storms, accidents to transmission lines, and station troubles. There have been complaints also of fluctuation in line voltage due to overloaded transmission lines which must be taken into consideration also.

These matters may not prove to be serious factors in the majority of industries, although they might give rise to comment in some of our industrial power plants which claim freedom from shutdown for periods of many years.

While it is the usual thing to eliminate the possibility of protracted shutdown periods as far as possible by the use of duplicate transmission lines, nevertheless the matter is important enough to deserve most careful consideration by central-station engineers, as the possibility of interruptions to the service, however brief or infrequent, is often the deciding factor regardless of other elements.

CONCLUSION

In conclusion, it might be stated again that these paragraphs were written with the thought that they might be helpful to the great number of our smaller industrial plants, that they might let a little light into that corner of the factory which is too apt to be shrouded in mystery to the non-technical man in the office. The endeavor has been made to show what other industrial plants are actually doing in the way of power costs and to indicate how good operation can be known and poor operation remedied. Some of the earmarks of both good and poor operation have been pointed out so that when the manager steps into his boiler room or engine department he will perhaps be able to view what is going on with better understanding, and when he questions his engineer as to power conditions he may do so with more confidence. It has been suggested that before enlarged capacity, changes in machinery, or possibility of turning the plant over to the public-service corporation are discussed, it should first be ascertained that the present plant is operating at maximum efficiency, using up-to-date methods. It has been also suggested that the day of the old-line steam engineer who knows nothing of modern methods of figuring and operating is well past, and that many such can be supplemented profitably with operators of better understanding while giving every consideration to the older men. Examples of modern power-plant reports have been shown with explanations of the function they serve and of their value as a basis of comparison from month to month and between different plants. It has been shown that such power reports, to be of value, must give definite cost data which will be comparable and which will help the manager in making up his estimates, and in making power-plant studies, and will tell him whether the tendency is toward improved operation or otherwise. It has been suggested that no operating or power data are of value unless the instruments used in the power house for collecting the facts are correct. Few such instruments will remain accurate for any length of time, and all must be checked and calibrated with regularity and frequency.

Certain charts and curves that may be used as aids to economy and indicators of operating conditions are shown. Such schemes are valuable in that they picture conditions in so simple a way that the meaning is readily grasped, yet they help both the manager and the engineer to maintain the plant on the highest possible plane of efficiency.

Some little space is given to the matter of public-service electric power. It has been suggested that the advantages of this service should be given careful unbiased consideration in many places, that power generation is a high-grade specialty which every industry cannot undertake successfully, and that the operation of private plants in some cases is not justified.

The Making of Special Slide Rules

Details of Steps to be Taken in Constructing Arithmetical and Logarithmic Scales, and of Their Use in the Construction of Slide Rules—Descriptions of Slide Rules for the Solution of Various Formulas

By G. W. GREENWOOD,[1] UNIONTOWN, PA.

MAN has always had recourse to mechanical aid in calculations. His "first aid" in computations were his fingers; and these have left their impress, doubtless for all time, in our method of counting by tens, although the old Babylonian method of counting by the dozen would have advantages over the decimal system. To our fingers is also due the name applied to the ten symbols of our system of numeration, the word "digit" being the Latin word for "finger."

Another ancient aid to computation were "counters" of various kinds. The word "calculus" is the Latin word for "pebble," and to "calculate" in its original sense meant to compute by the use of pebbles. This manner of mechanical computation still exists in various forms of the abacus. This instrument, used principally in adding and subtracting, was apparently invented independently among different races. It is still in current commercial use in some countries, as well as in the primary departments of our schools.

The development of a single individual along accounting lines follows the history of mankind. He usually begins with his fingers and follows by using pebbles or other counters. Work executed in a factory is frequently "tallied" by means of strokes, every fifth of which crosses the preceding four diagonally.

A few hundred years ago multiplication was considered quite difficult, while division was a feat essayed only by the most skilled mathematicians. Napier devised a system of rods, facetiously called "Napier's bones," as an aid to multiplication. At about the same time he invented logarithms. The importance of logarithms is too well known to merit more than mere mention.

Books of logarithms belong to another system of aids to computation, namely, mathematical tables. There are tables of squares, cubes, square roots, cube roots, and of many other functions of numbers. The preparation of special tables of a certain class is described in Management's Handbook.

Other valuable artificial aids are graphs, especially for functions of a single variable, such as simple curves and alignment charts. Then there are many forms of computing machines for multiplying, dividing, integrating, etc.

Following close upon the invention of logarithms came the invention of the slide rule. In addition to articles in encyclopedias, there are excellent histories of slide rules available.

The common slide rule consists of three scales, two fixed, the third sliding between them. In its simplest form it was used for multiplying and dividing, but by adding different scales slide rules have been made to solve an endless variety of problems.

Where one has continually to apply the same formula, involving products or quotients of functions of different variables, it is quite easy to make a slide rule which will give the desired results easily and quickly.

To make a slide rule requires no more ability than is required to work the problem itself. The material required is merely a few strips of tongued and grooved wood which will not warp; a book of logarithms, and a rule divided into tenths of an inch. It requires no special mathematical ability. In fact, it is far more easy to make a slide rule than it is to draw to scale a simple piece of machinery.

And then, after it is made, it requires little more intelligence to handle such a special slide rule than is required to handle a tape measure. By its use results have been secured in scientific management which would otherwise have not been possible.

Sometime ago the author had occasion to investigate the construction of special slide rules. Many descriptions and illustrations of such rules were found, but, so far as it was possible to discover, there is no published description giving in detail the method of constructing such rules. For this reason the present paper was written.

In the following description the steps are given in considerable detail, with frequent repetitions so as to avoid back references. The first section deals with arithmetical scales and the second with logarithmic scales, and each illustration is one used later in the construction of slide rules. Then follow descriptions of slide rules for the solution of different formulas.

Arithmetical Scales

§2 An arithmetical scale is one on which the distance between any two points bears a constant ratio to the difference between the two corresponding numbers. The constant ratio is called the "modulus." Thus if the scale of a map is 5 miles to the inch, the distance in inches between any two points is 0.2 times the difference between the numbers of miles corresponding to these points.

§3 Suppose it is desired to construct an arithmetical scale 8 in. long for numbers ranging from 0.22006 to 1.06160, both inclusive. The first step in locating intermediate points is to find the modulus. Since we are given two points 8 in. apart together with the numbers corresponding to these points, the modulus is found by obtaining the ratio of this distance to the difference between the corresponding numbers; thus,

$$\text{Modulus} = \frac{8}{1.06160 - 0.22006} = \frac{8}{0.84154} = 9.51$$

Assume that the scale reads from left to right, the smaller number being at the left. The position of any intermediate point is found thus: From the number corresponding to the point deduct 0.22006. Multiply this difference by the modulus, 9.51. The product is the distance, in inches, of the point from the left end of the scale. Stated algebraically, if h denotes the distance of any point g from the left end, g being the number corresponding to the point,

$$h = 9.51\ (g - 0.22006)$$

Suppose it is required to locate points corresponding to the numbers 0.30103, 0.47712, 0.60206, 0.69897, 0.77815, 0.84510, 0.90309, 0.95424, and 1.00000. The simplest plan is first to set down these numbers in a column, subtract 0.22006 from each, set down the remainders in another column, and in a third column place the products of these differences by 9.51; thus,

g	$g - 0.22006$	h
0.30103	0.08097	0.77
0.47712	0.25706	2.44
0.60206	0.38200	3.63
0.69897	0.47891	4.55
0.77815	0.55809	5.31
0.84510	0.62504	5.94
0.90309	0.68303	6.50
0.95424	0.73418	6.98
1.00000	0.77994	7.42

As a check on the accuracy of the work, take the total of the first column, subtract 0.22006 nine times (the number of items in the column), and see if the remainder is the same as the total of the second column. Then multiply this second total by 9.51 and see if the product differs from the total of the third column by less than nine times 0.005 (0.005 being the limit of error in each of the items in the last column).

Thus the total of the first column is 6.55976. Nine times 0.22006 is 1.98054. Deducting this leaves 4.57922, which is the same as the total of the second column. Multiplying by 9.51 gives 43.55, and the total of the last column is 43.54.

Next take a scale divided into inches and tenths of an inch and lay off from the left end the distances shown in the third column,

[1] Warren Bldg., N. Mt. Vernon Ave.

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designating the points by the corresponding numbers in the first column (Fig. 1).

Instead of using a decimally divided scale, good results can be obtained by using a strip of ordinary cross-section paper divided into inches and tenths of an inch, but care must be taken to see that the paper is ruled uniformly; sometimes it is not.

§4 Numbers corresponding to points on a scale may be negative as well as positive. Suppose a scale 8 in. long is to be constructed for a series of numbers from —0.03530 to —0.87686, both inclusive. Since the numbers are negative, the one whose absolute magnitude is greater is taken as the left end. The modulus is found by taking

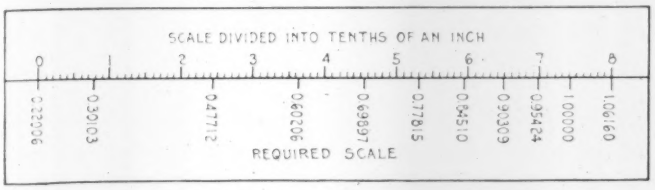

FIG. 1

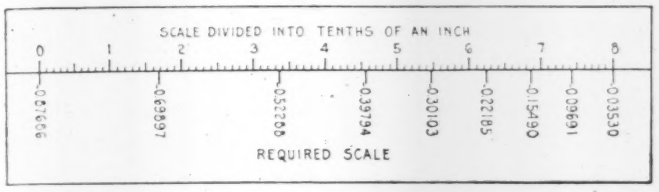

FIG. 2

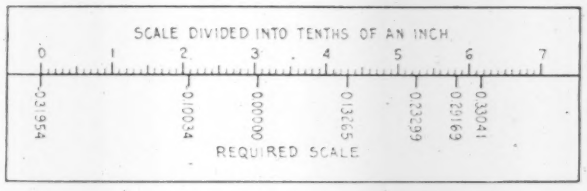

FIG. 3

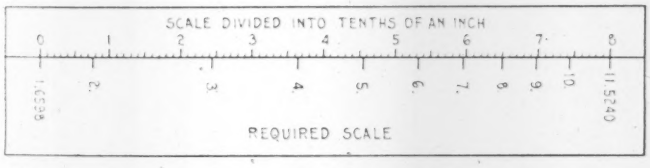

FIG. 4

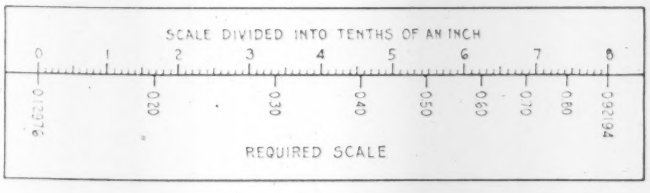

FIG. 5

the ratio of the distance between these two numbers to their difference; thus,

$$\text{Modulus} = \frac{8}{-0.03530 - (-0.87686)} = \frac{8}{0.84156} = 9.51$$

The position of any intermediate point is found by deducting —0.87686 from the number corresponding to the point (that is, by adding 0.87686 to the number) and multiplying the result by 9.51. The product is the distance, in inches, of the point from the left end. Stated algebraically, the distance h of any point g from the left end is given by

$$h = 9.51\ (g + 0.87686)$$

Suppose the points that are to be located correspond to —0.69897, —0.52288, —0.39794, —0.30103, —0.22185, —0.15490, and —0.09691. Construct a table as follows:

g	$g + 0.87686$	h
—0.69897	0.17789	1.69
—0.52288	0.35398	3.37
—0.39794	0.47892	4.56
—0.30103	0.57583	5.48
—0.22185	0.65501	6.23
—0.15490	0.72196	6.87
—0.09691	0.77995	7.42

The length will be found to be 8 in. The resultant scale is shown in Fig. 2.

§5 Sometimes the initial and final values of a series of numbers are given, but the modulus is given instead of the length. Consider a scale of numbers from —0.31954 to +0.33041, both inclusive, the series changing from negative to positive numbers. Suppose the intermediate points to be located correspond to —0.10034, 0.00000, 0.13265, 0.23299, 0.29169, and that instead of being given the length, we are given 9.51 as the modulus. The table required is as follows, the figures in the last column being 9.51 times those in the second column:

g	$g + 0.31954$	h
—0.10034	0.21920	2.09
0.00000	0.31954	3.04
0.13265	0.45219	4.30
0.23299	0.55253	5.26
0.29169	0.61123	5.81

The length of the scale is 9.51(0.33041 + 0.31954), i.e., 9.51 × 0.64995 or 6.18 in. The resulting scale is shown in Fig. 3.

LOGARITHMIC SCALES

§6 A logarithmic scale is one in which the distance between any two points bears a constant ratio to the difference between the *logarithms* of the corresponding numbers, or to the difference between the *logarithms of a definite function* of these numbers.

In other words, to construct a logarithmic scale, one takes the logarithms of the maximum and minimum values of the series of numbers (or of functions) together with the logarithms of any intermediate values which it is desired to designate, and then constructs an *arithmetical* scale for these *logarithms;* but the numbers themselves (not their logarithms) are annexed to the points so determined.

§7 Suppose it is desired to construct a logarithmic scale 8 in. long for numbers ranging from 1.6598 to 11.5240, both inclusive. The first step is to find the logarithms of these terminal values, and from these to determine the modulus. Thus,

$$\text{Modulus} = \frac{8}{\log 11.5240 - \log 1.6598} = \frac{8}{1.06160 - 0.22006} = 9.51$$

We are therefore to construct an arithmetical scale 8 in. long for numbers ranging from 0.22006 to 1.06160, both inclusive, and have found the modulus to be 9.51.

To locate any intermediate point G, first find log G, deduct 0.22006, and then multiply by 9.51. The result is the distance, in inches, from the left end of the point corresponding to the number G. Stated algebraically, if h denotes the distance in inches from the left end of a point corresponding to the number G, we have

$$h = 9.51\ (\log G - 0.22006)$$

Suppose the values of G to be designated on the scale are 2, 3, 4, 5, 6, 7, 8, 9, and 10. For convenience a table is prepared in which these numbers appear in the first column; in the second are their logarithms; and in the third are the differences between these logarithms and the logarithm of the initial number of the series (corresponding to the left end of the scale); in the fourth column are the products of these differences by 9.51, which converts these differences into distances in inches. The required table is as follows:

G	$\log G$	$\log G - 0.22006$	h
2	0.30103	0.08097	0.77
3	0.47712	0.25706	2.44
4	0.60206	0.38200	3.63
5	0.69897	0.47891	4.55
6	0.77815	0.55809	5.31
7	0.84510	0.62504	5.94
8	0.90309	0.68303	6.50
9	0.95424	0.73418	6.98
10	1.00000	0.77994	7.42

Using a scale divided into inches and tenths of an inch, the distances shown in the right-hand column are laid off and the points so found are designated by the numbers shown in the first column. Or, a strip of cross-section paper divided into tenths of an inch may be used instead, provided the paper is ruled with sufficient accuracy. The scale is shown in Fig. 4.

§8 Next consider a scale for numbers less than 1, so that the logarithms are negative; for instance, a scale 8 in. long for values of G from 0.13278 to 0.92194, both inclusive. The modulus is determined by taking the ratio of the length to the difference between the logarithms of the maximum and minimum values of G. Thus,

$$\text{Modulus} = \frac{8}{\log 0.92194 - \log 0.13278}$$

$$= \frac{8}{(9.96470 - 10) - (9.12314 - 10)} = \frac{8}{0.84156} = 9.51$$

We therefore construct an arithmetical scale whose modulus is 9.51 for numbers ranging from $(9.12314 - 10)$ to $(9.96470 - 10)$, i.e., from -0.87686 to -0.03530.

To locate any intermediate point G, find $\log G$, deduct -0.87686 (i.e., add 0.87686), and multiply by 9.51. The result is the distance in inches of the point from the left end of the scale. Stated algebraically, if h is the distance in inches from the left end of the point corresponding to G,

$$h = 9.51 (\log G + 0.87686)$$

Suppose the values of G to be designated are 0.20, 0.30, 0.40, 0.50, 0.60, 0.70, and 0.80. A table is prepared as follows:

G	$\log G$	$\log G + 0.87686$	h
0.20	$9.30103 - 10$	0.17789	1.69
0.30	$9.47712 - 10$	0.35398	3.37
0.40	$9.60206 - 10$	0.47892	4.55
0.50	$9.69897 - 10$	0.57583	5.48
0.60	$9.77815 - 10$	0.65501	6.23
0.70	$9.84510 - 10$	0.72196	6.87
0.80	$9.90309 - 10$	0.77995	7.42

The scale is shown in Fig. 5.

§9 Next consider a scale in which a function of a number is to be used instead of the number, and suppose the modulus is given instead of the length. For instance, construct a scale for values of $A^{1/2}$ for values of A from 12 to 29, both inclusive, the modulus of the scale being 9.51. One takes the logarithms of the maximum and minimum values of $A^{1/2}$ together with the logarithms of any chosen intermediate values of $A^{1/2}$ and constructs an arithmetical scale for these numbers having for its modulus 9.51. The minimum and maximum values of $\log A^{1/2}$ are found as follows:

$$\log 12^{1/2} = \frac{1}{2} \log 12 = \frac{1}{2} \times 1.07918 = 0.53959$$

$$\log 29^{1/2} = \frac{1}{2} \log 29 = \frac{1}{2} \times 1.46420 = 0.73120$$

The length of the scale is 9.51 times the difference between these two numbers, i.e., 9.51 (0.73120 — 0.53959) or 1.82.

Suppose the values of A to be shown are 15, 20, and 25. The required table is as follows and the scale is shown in Fig. 6:

A	$\log A^{1/2}$	$\log A^{1/2} - 0.53959$	h
15	0.58805	0.04846	0.46
20	0.65052	0.11093	1.05
25	0.69897	0.15938	1.52

§10 Next consider a scale for values of a function in which some of the values are less than 1, involving negative logarithms. For instance, $B^{1/3}$ for values of B from 0.11 to 9.80, both inclusive, whose modulus is 9.51. The logarithms of the minimum and maximum values of $B^{1/3}$ are found as follows:

$$\log 0.11^{1/3} = \frac{1}{3} \log 0.11 = \frac{1}{3} \times (9.04139 - 10)$$

$$= \frac{1}{3} \times (29.04139 - 30) = 9.68046 - 10$$

$$= -0.31954$$

$$\log 9.80^{1/3} = \frac{1}{3} \log 9.80 = \frac{1}{3} \times 0.99123 = 0.33041$$

The length of the scale is 9.51 (0.33041 + 0.31954) or 6.18 in., and the distance from the left end, in inches, for any intermediate value of B, is given by

$$h = 9.51 (\log B^{1/3} + 0.31954)$$

Suppose the values of B to be indicated are 0.5, 1.0, 2.5, 5.0, and 7.5. The required table is as follows, and the scale is shown in Fig. 7:

B	$\log B^{1/3}$	$\log B^{1/3} + 0.31954$	h
0.5	-0.10034	0.21920	2.09
1.0	0.00000	0.31954	3.04
2.5	0.13265	0.45219	4.30
5.0	0.23299	0.55253	5.26
7.5	0.29169	0.61123	5.81

§11 Sometimes a function increases while the variable decreases. In such a scale the numbers decrease from left to right. Consider a scale for $A^{-1/2}$ where A has values from 0.03343 to 0.08333, whose modulus is 9.51. The value of $A^{-1/2}$ is least when A is greatest,

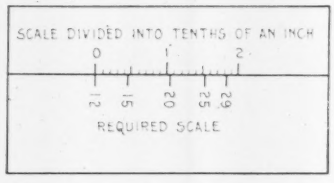

FIG. 6

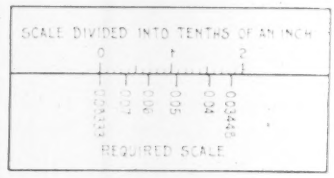

FIG. 8

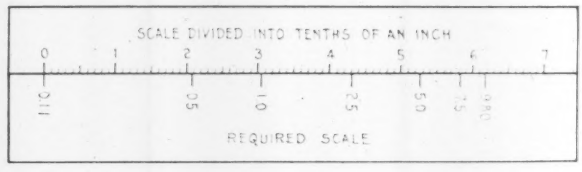

FIG. 7

i.e., when A is equal to 0.08333. The left end of the scale corresponding to the minimum value of $A^{-1/2}$ is given by

$$\log 0.08333^{-1/2} = -\frac{1}{2} \times \log 0.08333 = -\frac{1}{2} \times (8.92080 - 10)$$

$$= -\frac{1}{2} \times (18.92080 - 20) = -9.46040 + 10$$

$$= 0.53960$$

The right-hand end of the scale corresponding to the maximum value of $A^{-1/2}$ is given by

$$\log 0.03443^{-1/2} = -\frac{1}{2} \log 0.03443 = -\frac{1}{2} \times (8.53694 - 10)$$

$$= -\frac{1}{2} \times (18.53694 - 20) = -9.26847 + 10$$

$$= 0.73153$$

The length of the scale is 1.83.

The position of a point corresponding to an assigned value of A is given by

$$h = 9.51 (\log A^{-1/2} - 0.53960)$$

Assuming that the points to be designated are 0.07, 0.06, 0.05, and 0.04, a table is prepared as follows, and the scale is shown in Fig. 8:

A	$\log A^{-1/2}$	$\log A^{-1/2} - 0.53960$	h
0.07	0.57745	0.03785	0.36
0.06	0.61092	0.07132	0.68
0.05	0.65052	0.11092	1.06
0.04	0.69897	0.15937	1.52

SPECIAL SLIDE RULES

§12 Special slide rules are in general used to find the product of two or more functions (quotients being regarded as factors whose

exponents are negative). Logarithmic scales are constructed for each of the functions, the end points of each scale being the maximum and minimum values of the corresponding functions. The same modulus is used for all scales.

In addition to scales for the given functions, another one is prepared whose end points are the maximum and minimum values of the series of products.

The modulus may be chosen arbitrarily and the length of all scales figured from it, or the length of any one scale may be arbitrarily set and the modulus computed from this scale, the same modulus being then applied to the other scales. It is simpler to assign a length to the scale covering the product, find the modulus, and apply this modulus in computing the other scales. These points will be more clearly seen from the illustrations which follow.

§13 A type of equation most frequently solved by special slide rules is of the form

$$G = A^m B^n C^p \ldots .$$

where $A,B,C,\ldots$ are variables or functions of variables whose ranges of values are limited, and $m, n, p,\ldots$ are constants. The range of G depends naturally on the limiting values of the product on the right of the equation.

SLIDE RULE FOR FINDING $A^{1/2}B^{1/3}$

§14 Consider the equation $G = A^{1/2}B^{1/3}$, where the values of A range from 12 to 29, both inclusive, and the values of B range from 0.11 to 9.80, both inclusive.

Before considering how special slide rules are prepared, consider Fig. 9, which is intended for the solution of this equation. It consists of two scales, A and G, which are fixed, the left ends being in line, while between them slides a scale marked B whose left end carries an arrow pointing to the A-scale. Suppose it is desired to find the value of $20^{1/2} \times 2.5^{1/3}$. Slide the B-scale until the arrow points to 20 on the A-scale. Under 2.5 on the B-scale is found the result. It can be seen that the product is approximately equal to 6.1.

§15 The rule is prepared thus: Determine first the maximum and minimum values of G. The maximum value, given by

$$\begin{aligned}\log G &= \log 29^{1/2} + \log 9.80^{1/3}\\ &= \frac{1}{2}\log 29 + \frac{1}{3}\log 9.80\\ &= \frac{1}{2} \times 1.46240 + \frac{1}{3} \times 0.99123\\ &= 1.06161\end{aligned}$$

is 11.524. The minimum value, given by

$$\begin{aligned}\log G &= \log 12^{1/2} + \log 0.11^{1/3}\\ &= \frac{1}{2}\log 12 + \frac{1}{3}\log 0.11\\ &= \frac{1}{2} \times 1.07918 + \frac{1}{3} \times (29.04139 - 30)\\ &= 0.53959 + 9.68046 - 10\\ &= 0.22005\end{aligned}$$

is 1.6598. If the length of the scale is to be 8 in., our next step is to construct a logarithmic scale for the values of G from 1.6598 to 11.5240, both inclusive, whose length is 8 in. This has been done in Sec. 7, Fig. 4, the modulus being found to be 9.51. The divisions of the scale may be as fine or as coarse as one pleases.

The next step is to prepare logarithmic scales for values of $A^{1/2}$ where A has values from 12 to 29, both inclusive, the modulus of which is 9.51. This has been done in Sec. 9, Fig. 6.

Finally construct a logarithmic scale for values of $B^{1/3}$, where B has values from 0.11 to 9.80, both inclusive, whose modulus is 9.51. This has been done in Sec. 10, Fig. 7.

The G-scales and one of the others are then fixed with the left ends in line, the remaining scale sliding between these and having an arrow at the left end (Fig. 9). The method of using the slide to find values of G has already been described.

§16 The slide can also be used to find A or B when values are assigned to the other two letters. For example, if values of B and G are given, set the slide so that these are opposite each other, and opposite the arrow read the corresponding value of A on the A-scale. If values of A and G are given, set the arrow opposite the given value of A and opposite the given value of G will be found on the B-scale the required value of B.

§17 The A-scale could just as well have been placed on the slide, the B-scale being fixed, but in general there is less sliding required if the smaller of the two scales is taken as the fixed scale, as in this illustration.

§18 The A-scale (where this is the fixed scale) can be moved to the right provided the arrow on the B-scale is moved to the right the same distance. In Fig. 10 the A-scale is placed centrally over the G scale, merely for the sake of symmetry, by moving it to the right a distance equal to one-half the difference between its length (1.82 in.) and the length of the G-scale (8 in.), or a distance of 3.09 in., the arrow on the B-scale being moved the same distance to the right. Also, since the margin on each side of the B-scale is immaterial, these two margins are made the same so that when closed the end points of all three scales will be symmetrical. It will then be found not only that all the scales are symmetrically placed, but that the arrow is in the center also.

SLIDE RULE FOR FINDING $A^{-1/2}B^{1/3}$

§19 Consider a case in which one of the exponents is negative.

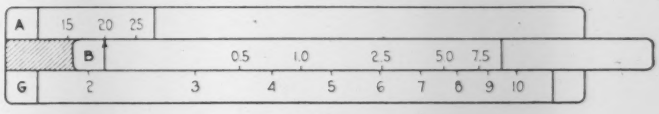

FIG. 9

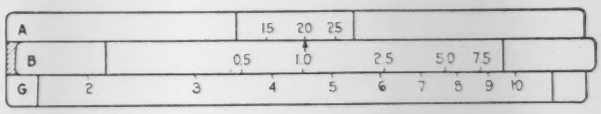

FIG. 10

For instance,

$$G = A^{-1/2}B^{1/3}$$

the limits of A being 0.03448 and 0.08333 and those of B being 0.11 and 9.80 as before.

The first step is to determine the limiting values of G. Since the exponent of A is negative, the value of $A^{-1/2}$ is greatest when the value of A is least. The maximum value of G, given by

$$\begin{aligned}\log G &= \log 0.03448^{-1/2} + \log 9.80^{1/3}\\ &= -\frac{1}{2}\log 0.03448 + \frac{1}{3}\log 9.80\\ &= -\frac{1}{2} \times (18.53757 - 20) + \frac{1}{3} \times 0.99123\\ &= -9.26870 + 10 + 0.33041\\ &= 1.06162\end{aligned}$$

is 11.524. The minimum value, given by

$$\begin{aligned}\log G &= \log 0.08333^{-1/2} + \log 0.11^{1/3}\\ &= -\frac{1}{2}\log 0.08333 + \frac{1}{3}\log 0.11\\ &= -\frac{1}{2} \times (18.92080 - 20)\\ &\qquad + \frac{1}{3} \times (29.04139 - 30)\\ &= -9.46040 + 10 + 9.68046 - 10\\ &= 0.22006\end{aligned}$$

is 1.6598. If the length of the scale is 8 in., it is the same as Fig. 4, Sec. 7, the modulus being 9.51.

Next prepare a logarithmic scale for values of $A^{-1/2}$ where A has values from 0.08333 to 0.03448, whose modulus is 9.51. This has been done in Sec. 10, Fig. 8.

The B-scale is the same as before. The completed scale is shown in Fig. 11, where it is set for $A = 0.06$, values of G being opposite corresponding values of B. For instance, it is seen that $0.06^{-1/2} \times 7.5^{1/2}$ is approximately equal to 8, the computed value being 7.99.

Slide Rule for Finding $KA^{1/2}B^{1/3}$

§20 Consider next an equation containing a constant factor. For example,

$$G = KA^{1/2}B^{1/3}$$

Suppose $K = 0.08$ and that the limiting values of A are 12 and 29, and of B, 0.11 and 9.80, the same as before.

The maximum value of G, given by

$$\begin{aligned}\log G &= \log 0.08 + \log 29^{1/2} + \log 9.80^{1/3} \\ &= (8.90309 - 10) + 0.73120 + 0.33041 \\ &= -0.03530\end{aligned}$$

is 0.92194. The minimum value, given by

$$\begin{aligned}\log G &= \log 0.08 + \log 12^{1/2} + \log 0.11^{1/3} \\ &= (8.90309 - 10) + 0.53959 + 9.68046 - 10 \\ &= -0.87686\end{aligned}$$

is 0.13278. These limiting values are, of course, 0.08 times those obtained in Sec. 15, but there is an advantage in handling each case by itself to a large extent.

For an 8-in. scale the modulus is found by taking the ratio of the length to the difference between the logarithms of the end values of G. Thus,

$$\text{Modulus} = \frac{8}{-0.03530 + 0.87686} = \frac{8}{0.84156} = 9.51$$

The modulus is the same as in Sec. 15, which is natural since $\log K$ is added to each value of $\log G$ and therefore does not affect the difference between any two values of $\log G$.

This G-scale has been worked out in Sec. 8, Fig. 5. The A- and B-scales are the same as in Fig. 10, the completed rule being shown in Fig. 12. It is seen in the figure that $0.08 \times 20^{1/2} \times 7.5^{1/3}$ is approximately equal to 0.70.

Slide Rule for Finding $A^{1/2}B^{1/3}C$

§21 Consider a rule for finding the product of three variable functions, such as $G = A^{1/2}B^{1/3}C$, where values of A range from 12 to 29; of B, from 13.75 to 153.13; and of C, from 0.20 to 0.40.

The minimum value of G, given by

$$\begin{aligned}\log G &= \log 12^{1/2} + \log 13.75^{1/3} + \log 0.20 \\ &= \frac{1}{2}\log 12 + \frac{1}{3}\log 13.75 + \log 0.20 \\ &= \frac{1}{2} \times 1.07918 + \frac{1}{3} \times 1.13830 + (9.30103 - 10) \\ &= 0.53959 + 0.37943 + 9.30103 - 10 \\ &= 0.22005\end{aligned}$$

is 1.6598. The maximum value given by

$$\begin{aligned}\log G &= \log 29^{1/2} + \log 153.13^{1/2} + \log 0.40 \\ &= \frac{1}{2}\log 29 + \frac{1}{3}\log 153.13 + \log 0.40 \\ &= \frac{1}{2} \times 1.46240 + \frac{1}{3} \times 2.18506 + (9.60206 - 10) \\ &= 0.73120 + 0.72835 + 9.60206 - 10 \\ &= 1.06161\end{aligned}$$

is 11.524. If the length of the scale is 8 in., the modulus is 9.51 and the scale is the same as in Fig. 5, Sec. 6.

Using this modulus, logarithmic scales are then made for values of $A^{1/2}$, $B^{1/3}$, and C, the completed scale being shown in Fig. 13, the A-scale and the G-scale being fixed, and the other two sliding between them.

Any one of the A-, B-, or C-scales could have been taken as the fixed scale, and the other two could have been placed between, in either order.

Each scale is symmetrically placed, the arrows being in the center of each. This is merely for the sake of appearance.

As shown in the figure, $20^{1/2} \times 60^{1/3} \times 0.40$ is approximately equal to 7.

§22 A simpler form of scale for the solution of this equation, with only one slide, is shown in Fig. 14. Here the B-scale is reversed, so that when given values of A and B are opposite each other the required value of G is opposite the given value of C. For instance, in the illustration, the scale is set for $A = 20$, $B = 60$, and opposite 0.40 on the C-scale is read the corresponding value of G, which is approximately 7.

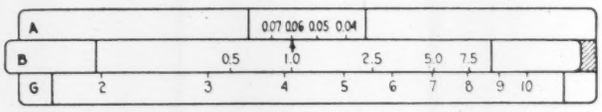

FIG. 11

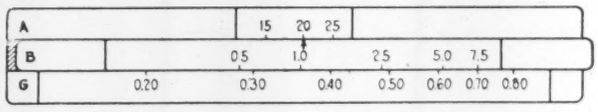

FIG. 12

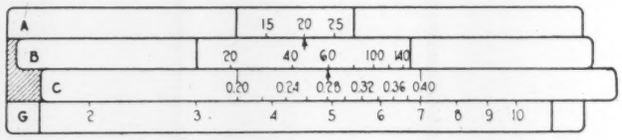

FIG. 13

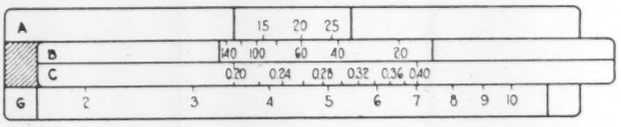

FIG. 14

Other Special Rules

§23 By alternately reversing the scales on the slides, it is possible to make a slide rule with two slides which will solve the equation

$$G = KA^mB^nC^pD^qE^r$$

where A, B, C, D, and E are independent variables, G is a dependent variable, and K, m, n, p, q, and r are constants. In locating the scales for such a rule which contains no arrows, they are simply placed so that each is in the center of the rule or the slide.

§24 If a runner is attached to the rule, it will make it easier to handle when estimating intermediate values which are not marked on the scales. It is not easy to judge the positions of two unmarked points simultaneously.

In preparing the illustrations, working models were first made using strips of paper on which the points designated were located by means of strips of cross-section paper divided into tenths of an inch. With these paper models it was found possible to determine values of G to one decimal place.

Small slide rules made of flat concentric disks, or of disks on a shaft, may be made in a similar manner, but they are not so easy to construct from a mechanical standpoint.

No attempt has been made here to cover exponential equations, nor equations involving addition and subtraction as well as multiplication and division.

The Gas Engines of the Maryland Plant of the Bethlehem Steel Company

Details of an Installation of Eighteen Large Engines Using Blast-Furnace Gas, Together with Data of Tests and Operating Records

By A. A. RAYMOND,[1] SPARROWS POINT, MD.

IN THIS plant, located at Sparrows Point, Md., there are eighteen gas engines operating on blast-furnace gas, twelve blowing engines and six engines driving electric generators. These eighteen units are housed in three buildings, connected to permit easy transfer of material. Each building is equipped with an overhead electric crane, two of them being of fifty tons capacity and one of forty tons.

The blowing engines are two-cylinder four-cycle double-acting. The first five installed in 1914 have gas cylinders 45 in. in diameter and air cylinders 80 in. in diameter, and the seven added since have 47-in. gas cylinders and 84-in. air cylinders; the stroke of all the engines is 60 in.

The generating units each consist of a four-cylinder twin-tandem engine having cylinders 47 in. in diameter and 60 in. stroke directly connected to a 25-cycle 6600-volt generator running at 88.3 r.p.m. and rated at 4000 kw. at 80 per cent load factor.

The fuel gas after passing through the wet washers at the furnaces is carried to the Theisen washers, of which there are eight distributed in the three buildings, each having a capacity of 850,000 cu. ft. per hr. Each Theisen is driven by a 150-hp. motor running at 355 r.p.m. All the Theisens discharge into a common clean-gas main from which there is an engine supply line to each building. Connected to the clean-gas main is a gas holder having a capacity of 202,000 cu. ft., the flow of gas to and from the holder being regulated by an automatic valve operated by a gasometer. This arrangement permits the successful operation of blowing engines in a one-furnace plant, the holder supplying gas to the blowing engines for the short time that the blast is off the furnace at casting time. This plant operated during 1921 with but one furnace in blast and produced 58,000,000 kw-hr. with the gas-driven generators, with a maximum hourly output of 1400 kw.

The total water required for the engines amounts to 316,600 gal. per hr., of which approximately half is salt water used for gas cleaning, the blowing engines using 4800 gal. of cooling water per engine-hour and the generator units 15,000 gal. per engine-hour. After passing through the engines the water is cooled by spraying, the average temperature of the water to the engines being 85 deg. fahr. and the leaving temperature 145 deg. fahr.

For starting the engines, air at 200 lb. pressure supplied by motor-driven compressors is used. It is stored in fifteen tanks located in the basement, and supplied to all engines from a common line.

Details of Engine Construction

Cylinders. The engine cylinders are made of cast steel in either the one- or two-piece type. Steel is used in preference to cast iron for the following reasons:

1 Owing to its superior strength, the walls can be made thinner than when cast iron is used, which permits a more rapid transfer of heat and more uniform cooling of the metal throughout.

2 The limit of thickness is not strength, but that necessary to make a good casting. This is especially of advantage in large dimensions where the thickness, if cast iron were used, would be prohibitive.

3 Steel is a better metal than cast iron to withstand, without fatigue, shocks due to explosion and also to withstand temperature stresses.

4 Steel can be put in tension with safety.

5 Owing to permissibly thinner steel cylinders, the wall temperature can be maintained at a satisfactory point with jacket water at a temperature much hotter than that allowable with cast-iron cylinders. This increases the thermal efficiencies, especially in large cylinders.

6 The use of steel permits designs of valve ports and other details which decrease the overall length. Bethlehem engines as compared to other makes of gas engines of the same horsepower are much shorter.

7 Steel permits a lighter engine and at the same time a much stronger one than cast iron.

Pistons. The pistons are of cast steel, cast in one piece without ribs. As far as possible the shape of the piston is spherical. The piston is thus safe against temperature stresses and rigid against explosive pressures. Having comparatively thin walls as compared to cast-iron pistons, there is better cooling. The weight is less than that of a cast-iron piston and there is consequently less piston-rod deflection. This also decreases the weight of the reciprocating parts, thus allowing speeds higher than would otherwise be safe. The packing rings are very deep, which prevents wear on the sides of the grooves. The piston is pressed on the rod and held in place by a nut which is doweled and staked.

The piston rod is bored for water cooling from both ends to avoid using any plugs. The water being introduced into and taken from the ends of the rod, it is unnecessary to drill transverse holes in the ends, thus avoiding any weakening of the rod at these points. The holes for the water inlet and outlet from the piston are drilled where the rod is largest in diameter and at an angle, so that a section through the rod where the holes are drilled is no weaker than any other section. Both rods are alike and are thus interchangeable by turning them end for end.

The piston-rod packing is designed with rings of the segmental type, floating on the piston rod. They are kept in place by springs having physical properties adapted to resist the effect of heat. There is no complication due to water cooling. The lubrication is excellent as the rod is oiled by droppers from the top. The packing is easily cleaned and inspected. All packing rings are made of cast iron.

Lubrication. A continuous-pressure oil system is used wherever possible. Each engine has its own oil pump driven from the end of the layshaft. Engine oil is supplied under pressure to the main bearings, crankpins and crosshead pins, and to the main and intermediate crosshead slides. From these the oil flows to the filters. Each engine having its own filter and pump, it is independent of any other engine in the power house.

Cylinder oil is supplied to the cylinders, piston packing and exhaust-valve stems by sight-feed oil pumps which are also driven from the layshaft. There are two inlets at each end of each cylinder, thus insuring a good and even distribution of oil on the piston rings. The time of injection of the oil is such as to thoroughly lubricate the cylinder before each explosion stroke.

Governing. The governor, of the standard spring-loaded type, is driven by gears. It does not act directly on the mixing valves but controls an oil-operated piston by means of a relay valve in hydraulic balance. The work required of the governor is constant regardless of the load of the engine or the physical condition of the mixing valves. This arrangement secures more rapid control under all operating conditions, which is extremely important for electric generators operating in parallel.

The mixing valve gives a mixture of absolutely constant quality under all loads. The amount of mixture is varied with the load, and at no time is the amount of gas so small as not to form a combustible mixture. By having the ports cut to 45 deg. an effect similar to a cut-off valve is obtained, yet at the same time the valve action is positive. The same design of valve is used for all gases.

[1] Chief Engineer, Gas Engine Department.

Contributed by the Oil and Gas Power Division for presentation at the Annual Meeting, New York, November 30 to December 4, 1925, of The American Society of Mechanical Engineers. All papers are subject to revision.

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The mixture is ignited by three spark plugs on each cylinder, of the make-and-break type, mechanically driven. Current is supplied at 110 volts d.c. Induction coils are used to increase the density of the spark.

TABLE 1 DATA OF TESTS AND OPERATING RECORDS OF GAS ENGINES AT THE MARYLAND PLANT OF THE BETHLEHEM STEEL CO.

Blowing Engines	
Average free air per minute required per furnace, cu. ft.	42,000
Average blast pressure, lb. gage	20
Engines per furnace	2
Revolutions per minute per engine, 80-in. air tub	69
Revolutions per minute per engine, 84-in. air tub	62
B.t.u. per cu. ft. of dry fuel gas	105
Cu. ft. gas consumed per engine-hour	145,000
Total heat consumed per engine-hour, B.t.u.	15,244,000
Thermal efficiency per i.hp. per hour in gas cylinders, per cent	28
Heat Balance	
Heat carried off by cooling water, per cent	27.4
Heat in exhaust gas, per cent	36.7
Heat unaccounted for, per cent	7.9
Heat converted into work in cylinders, per cent	28.0
Total hourly gas consumption of blowing engines, cu. ft.	1,745,000
Generator-Unit Engines	
Kilowatts per unit per hour	3400
B.t.u. per cu. ft. of dry gas	105
Cu. ft. of gas consumed per engine-hour	481,000
Cu. ft. of gas consumed per kw-hr.	143
Total heat to engine per hour, B.t.u.	50,524,430
Heat Balance	
Heat carried off by cooling water, per cent	28.0
Heat carried off by exhaust gas, per cent	38.0
Heat lost and unaccounted for, per cent	5.4
Heat converted into work in the cylinders, per cent	28.6
Total hourly consumption of generator-unit engines, cu. ft.	3,000,000
Total hourly gas consumption of blowing and generator-unit engines, cu. ft.	4,745,000

Results of Tests and Operating Records

Tables 1 and 2 are the results of tests and operating records.

Of the total heat available from the exhaust gases the amount now utilized is very small. Besides the hot water furnished to the boilers from the circulation system, sufficient low-pressure steam has been generated during the winter months to supply heating coils in the oil filters and cylinder-oil pumps and furnish some heat in the buildings. This steam is generated by the use of heating coils placed in the exhaust headers, the coils being supplied with water leaving the engines at about 145 deg. fahr.

Total steam generated, boiler hp-hr. per month	57,000
Total steam generated, boiler hp-hr. per engine-hour	19

Table 2 shows the total heat available from exhaust gases.

TABLE 2 TOTAL HEAT AVAILABLE FROM EXHAUST GASES OF BLOWING AND GENERATOR-UNIT ENGINES

Blowing Engines	
Total heat to engines per hour, B.t.u.	15,244,000
Total heat in wet exhaust gas per hour, B.t.u.	6,000,000
Temperature of exhaust gases, deg. fahr.	900
Heat convertible in waste-heat boilers, B.t.u. per h.	3,000,000
Boiler hp. available per hour	88.2
Generator-Unit Engines	
Total heat to engines per hour, B.t.u.	50,524,000
Total heat in wet exhaust gas per hour, B.t.u.	19,000,000
Temperature of exhaust gases, deg. fahr.	1,000
Heat convertible in waste-heat boilers, B.t.u. per h.	9,500,000
Boiler hp. available per hour	279.4
Exhaust heat from generator-unit engines utilized, per cent	6.5

The Question Mark in Machine Design

Difficulties Encountered in Analysis of Design Problems That Call for Exercise of Judgment—Loose Thinking and Its Probable Effects on Engineering Design—False Impressions Current with Regard to the Design of Machine Foundations, Bearings, Open-Side Machine Frames, Etc.

By FORREST E. CARDULLO,[1] CINCINNATI, OHIO

ALL PROBLEMS arising in the design, construction, and use of machinery ought to be solved by a rigorous and thorough analysis of the engineering elements involved. However, every engineer who has undertaken to solve very many such problems knows that it is often impossible to express his problem as a mathematical equation or series of equations. If, in order to escape this difficulty, the engineer has recourse to a more general process of reasoning, he finds it equally impossible to assign proper relative importance to elements which usually have no apparent relation, and which are often in serious conflict. He is compelled to choose between trains of analysis which lead to different conclusions, because they are based on different sets of premises, and his difficulty lies in the fact that he knows both sets of premises have in them large elements of truth. The engineer is therefore driven to supplement his analysis by the use of judgment, deciding each knotty element of his problem in accordance with the aggregate of the impressions relative thereto produced in his mind by his past experience. The difficulties just outlined arise from four sources.

First, the mathematical treatment required to solve the problem may be too difficult to permit of a solution by the particular engineer, or even by any present-day mathematician. For example, a designer may be obliged to give a complicated form to a part subject to known loads, and because of this complicated form the stresses and deflections produced do not permit of computation.

Second, the conditions of service may not be predictable. No one can tell what feed, depth of cut, and sharpness of tool a workman may use when he operates a machine tool, nor the hardness of the material cut. The assumptions of the designer are guesses pure and simple, and he cannot give a satisfactory reason for them. No bridge engineer can tell how hard the wind will blow, nor can any architect tell how much a snow load will be, nor how heavily a floor will be loaded.

Third, the requirements of service may not be predictable. Who can say that a deflection of 0.005 in. in a machine member is permissible, while a deflection of 0.007 in. is too great? By what process of reasoning does an automobile designer arrive at the conclusion that a level still-air speed of 60 miles per hour is just right, but that one of 55 miles is too low and one of 65 miles too high?

Fourth, the conditions of service may be fully known and the elements may be calculable, but the correct answer for one set of conditions may be entirely wrong for another set. For example, consider a shaft which is required to run at speeds of both 50 r.p.m. and of 1000 r.p.m. The proper oil for one speed is entirely incorrect for the other speed. Also the best running clearance for 1000 r.p.m. is entirely too much for 50 r.p.m.

Basing Designs on Guesswork

As a result of such difficulties, the designer is frequently obliged to base his design entirely on judgment. Two designers of equal ability will produce two entirely different designs, and if both are candid, each will have to admit that both designs are equally satisfactory, and furthermore, that neither design is what he would like it to be.

Because of these facts, many who are incapable of attempting an analysis of engineering problems believe they are justified in solving such problems by the use of judgment. They do not follow the methods of the engineer, who makes limiting assumptions and then by various trains of reasoning arrives at limits between which his answer must lie, and then chooses his answer within those limits by exercising a judgment in which only those experiences which bear on the problem and which are known to be in accord with the laws of science have any weight. Their procedure is to collect all the impressions which are suggested to their minds in connection with the problem by any mental process whatever, whether it be association of ideas, similarity of words, the likeness of function, or what not, and then out of this heterogeneous hodgepodge of impressions to arrive at a conclusion which may be, and often is, far from correct. To put it plainly, they substitute guesswork for analysis, and chance for judgment.

If engineers had to deal only with one another they would seldom encounter trouble from such loose habits of thought, but since engineering products must be sold to and used by people incapable of engineering analysis and unused to logical thought, the engineer, and particularly the sales engineer, must take account of them. In fact, the sales engineer is often willing to go further than necessary and request changes in design which have no engineering justification and sometimes are disadvantageous because he believes that they will appeal to a majority of his customers. In a similar spirit he will attribute to a good design of his own firm, virtues which it does not have, and to an equally good design of his competitors, faults which it does not possess. Having developed a line of sales argument having a real but very slight engineering foundation, he often advocates to the engineer that new designs should be developed primarily from the standpoint of his sales argument.

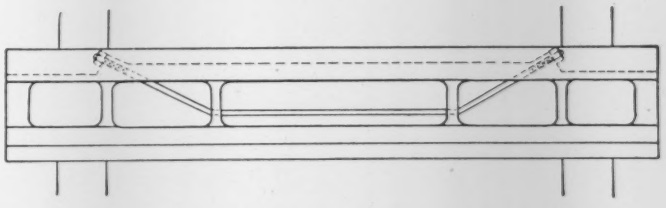

Fig. 1 Trussed Cross-Rail of Planer Type of Grinder

This peculiar attitude of mind is engendered because cumulative experience shows him that engineering products must be designed with an eye to sales as well as to service.

Loose Thinking and Its Probable Effects on Engineering Design

In order to discover the extent of such loose habits of thought, and of their probable effects on engineering design, the author formulated a hypothetical design problem and asked the opinions of a number of persons, engineers and others, all of whom had had mechanical experience, regarding a possible solution. This problem was as follows:

The cross-rail of a planer carries a head for mounting the cutting tool which weighs several hundred pounds. Ordinarily this weight is not sufficient to deflect the rail a measurable amount, so that the work will be sensibly flat when the finishing cut is taken. However, if the length of the rail or the weight of the head be greatly increased, as, for instance, in designing a planer type of grinder for very wide surfaces, this is no longer true, and the problem is presented of guiding the wheel in a straight line in spite of the deflection in the rail produced by the moving load. One solution of the difficulty is to truss the rail in the manner illustrated in Fig. 1, putting an initial tension in the truss rod just sufficient to keep the path of the wheel sensibly straight as the heavy head passes across the rail.

When the matter was presented to them the engineers all admitted that this was an effective solution of the problem, but were mostly of the opinion that prospective customers would regard the device

[1] Chief Engineer, G. A. Gray Company. Mem. A.S.M.E.

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as a bungling attempt to remedy a defect which ought not to exist, and suggested concealing the truss rod as a satisfactory solution of the problem of sales resistance. The machine users did, without exception, take exactly that view of the proposed device. They all demanded to know why the rail should not be made stiff enough so that there would be no deflection. Some of them refused to believe that this was not possible, *and maintained that any deflection in a machine-tool member was due to "lack of rigidity."* Further questioning elicited the idea that rigidity meant absolute immobility, and that the lack of this quality was due to some defect in the design, material, or workmanship of the structure. Several of those interviewed stated that they preferred the rail without such a device, because it would strain the metal, not realizing that the strains produced would be equal and opposite to those imposed by the weight of the moving head, and therefore would leave the rail casting free from strain under the conditions of service.

THE QUESTION OF MACHINE FOUNDATIONS

More often, however, the engineer encounters the effects of such loose thinking in the erection and use of engineering products. One of the things which gives a great deal of trouble to some machinery builders is the attitude of the average machinery user on the subject of foundations and on the method of attaching machinery to foundations. Machine frames are of two types: namely, those which have inherent rigidity and so have need of support at three points only; and those without inherent rigidity, which must be supported at intervals in order to preserve their form and maintain perfection of alignment. The average machine user attempts to confer rigidity on machines of the latter type by bolting or grouting them firmly to a foundation. Such a procedure is fatal to the satisfactory use of planers, long lathes, and other machines of that type.

The bed of such a machine should be set on a good foundation, but should never under any circumstances be bolted or grouted to that foundation. Instead, it should be supported on suitable wedges or other leveling devices at intervals of three to five feet. Many shop men, without stopping to analyze the conditions, assume that a foundation will remain true, and that if a machine bed without inherent rigidity be grouted to the foundation, it will add to the stiffness and strength of the bed and eliminate vibration. As a matter of fact, foundations sometimes settle from a quarter to half an inch, and the author has known of planer beds which were forcibly sprung over a quarter of an inch by being grouted firmly to a foundation which had settled. As a result the ways were no longer straight, the table vees touched only at the high spots, and the bearing surfaces were soon destroyed. Had this machine been set on leveling blocks placed at about four-foot intervals and the leveling blocks drawn up as often as necessary to compensate for the settling of the foundation, the planer would have remained accurate and the bearing surfaces perfect for the life of the machine.

This brings up the question of machine foundations in general. Machine foundations may be divided into rock-supported foundations, pile-supported foundations, and floating foundations. Rock-supported foundations are concrete or masonry structures which rest on rock or hard clay of such bearing power that the foundation does not settle measurably under the load of the machinery. Such a foundation may be in one piece or it may be a series of piers. If such a foundation is obtainable at a reasonable cost, it is the best sort of machine foundation, but even a rock-supported foundation is subject to some seasonal movement, and long beds or frames must not be grouted or bolted to it.

Where the foundation cannot be carried down to the rock because of the cost or for other reasons, it may be laid upon piles, columns, or beams which are rock-supported. Such a foundation is less subject to seasonal movement, but is more likely to settle at points where it carries concentrated loads.

A floating foundation is one laid on an ordinary earth surface which has been properly leveled and compacted. The foundation must be stiff enough so that it transmits the weight equally to all parts of the surface, and large enough so that the distributed load does not exceed the safe bearing power of the earth. It is usually in the form of a properly designed reinforced-concrete slab. Many persons mistakenly imagine that the supporting power of a floating foundation is proportional to its depth, and spend money unnecessarily for excavation and concrete, or, worse yet, erect machinery on floating piers. Obviously even a good floating foundation may settle, and provision must be made for keeping the machinery level. If the foundation carries only fixed weights, it can support a number of machines with only slight seasonal changes and very little settling. If, however, the loading varies from time to time, as, for instance, if the slab supports a column supporting a traveling crane, independent slabs should be provided for machines without inherent rigidity, while a common slab will do for a number of machines whose frames have inherent rigidity.

When a floating foundation is laid near a pile or rock-supported foundation, many masons think that they add to the rigidity and supporting power of the floating foundation by anchoring it at this point to the pile or rock-supported foundation. As a matter of fact, the foundation is then not as good as it would be if it were free from such support, for if the earth settles ever so slightly the foundation will no longer be true; while if the rock foundation be subject to moving loads, the floating slab will continually vary in its lever.

Civil engineers have given a great deal of study to the subject of foundations, and practically all civil-engineering structures are properly supported; but machinery users, instead of designing their foundations in accordance with the principles developed by civil engineers, usually design them by guess.

FALSE IMPRESSIONS CURRENT WITH REGARD TO THE DESIGN AND USE OF BEARINGS

The bearing is a machine element which has long been in use. Because of the great number of bearings in use, their importance, and their apparently erratic behavior, bearing design has received much thought, and almost every user of machinery has had considerable experience with bearing troubles. Consequently a great deal of misinformation and a great many false impressions are current with regard to the design and use of bearings.

The best bearing, where it is possible to use it, is a perfectly cylindrical shaft or pin revolving in a cylindrical hole, the hole being larger than the shaft by a sufficient amount to give an oil film of proper thickness. One of the things which is often done wrong is to "scrape in" or "fit" a bearing. "Scraping in" is the old-fashioned mechanical method of making a hole of the same size as the shaft. Therefore a bearing that has been scraped in has no room between the shaft and box for an oil film. Such a bearing invariably runs hot until it has been "run in," that is, until the box has been worn sufficiently large, or the shaft sufficiently small, to allow for an oil film. Since such a bearing does not usually wear to the proper form in running in, it wears too much and becomes loose. The designer attempts to remedy one mistake by another and divides his bearing so as to form two half-boxes, and provides shims which may be removed for closing in as the bearing wears. Half-boxes are often necessary in order to assemble the bearing, but in such a case the two half-boxes should be machined to a perfect cylindrical hole, and no shims should be used.

While a half-box may be necessary for reasons of assembly, a quarter-box is a mechanical crime. The purpose of a quarter-box may be to provide means for taking up the wear of the bearing, or it may be to provide adjustment by which the shaft may be aligned. It is just as easy to provide adjustment for a whole box or a half-box as it is for a quarter-box. Quarter-boxes wear rapidly because they cannot possibly be lubricated. The load either comes on one of the quarters, which gives an angle of only 90 deg. for support, or else it comes on two quarters and rapidly wears down the point of division between the parts of the box. The cracks at the points of division permit the oil to escape, and prevent the proper establishment of the oil film. In addition to being mechanically incorrect, a quarter-box is a very expensive form of bearing, and any designer will find that if he substitutes for it a properly designed half-box he will eliminate a great deal of his bearing trouble.

Properly designed bearings which have an abundant and continuous supply of clean lubricating oil will show no signs of wear, unless they are required to function under extreme variations in speed. If lubrication is imperfect, or the design of the bearing is such as to produce local distortion, or the bearing operates over an extreme range of speeds, wear may occur. Wear usually takes the form of a cutting or scoring of the bearing surfaces. This may

be prevented by making the two rubbing materials of sufficiently dissimilar hardness so that the softest crystals in the harder material are harder than the hardest crystals in the softer material, and then giving the harder material a sufficiently true and smooth surface. The author is aware that this is contrary to the accepted theory of bearing metals, which requires that a bearing metal shall consist of a certain proportion of hard crystals embedded in a soft matrix. However, there is no theoretical reason for regarding a bearing metal of that type as satisfactory, except in comparison with materials which are obviously unfitted to serve as bearing metals. It is obvious, however, that a box and shaft meeting the specifications outlined above cannot cut and therefore cannot wear. It only remains to proportion the bearing area and choose the lubricant and clearance with due regard to the speed, so that the oil pressure shall at no point exceed the compression elastic limit of the soft material.

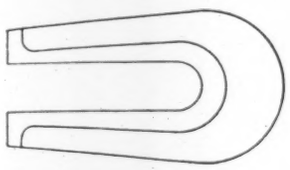

FIG. 2 CONVENTIONAL DESIGN OF OPEN-SIDE MACHINE FRAME

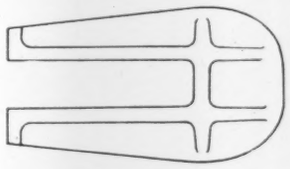

FIG. 3 PROPER METHOD OF DESIGNING AN OPEN-SIDE MACHINE FRAME

The author cannot in such a paper as this discuss the theory of lubrication of a cylindrical bearing, or the proper design and function of oil grooving. It must be said, however, that the figure eights, criss-crosses, and complicated systems of reversed spirals with which many bearings are provided show that their designers are woefully ignorant on these points.

There is one type of bearing concerning which there is no satisfactory theory of lubrication, namely, the oscillating bearing, where the external forces do not periodically reverse, or at least relieve themselves. Without such a theory we are obliged to grope in the dark in designing such a bearing, but that is no excuse for adopting the same procedure with other types of bearing where the light is available.

In certain types of bearings, such, for instance, as lathe-spindle bearings, the conditions of service are such as to preclude perfect lubrication at certain times. At the same time it is necessary to maintain correct alignment in spite of wear. Such bearings are often made slightly conical and provided with endwise adjustment to compensate for wear. However, the materials are usually so chosen that it is the box and not the spindle which wears, and since the wear is usually concentrated along one side of the box, alignment is not maintained. If the usual design were reversed and the box made of hardened steel while the revolving element were made of a softer and suitable material, the wear would be distributed around the entire circumference, and correct alignment would always be maintained.

Many design errors have the sanction of high engineering authority. One example of this is the case of the feed mechanisms of lathes, which are frequently over-powered as a result of Mr. Taylor's dictum that the feed mechanism should be capable of developing the same pressure at the tool point as the driving mechanism. The only excuse for this is that in case a tool loses its edge, the feeding pressure on the tool *may* equal the cutting pressure, and the feed gearing will break if it is not proportioned by the above rule. It is equally true that if certain commonly used tool forms lose their cutting edge, the feeding pressure may equal ten times the cutting pressure, and the feed gearing will break if proportioned by the above rule. The obvious remedy is to proportion the feed mechanism with an ample factor of safety against all reasonable conditions, and then to protect it by a reliable slipping clutch or other safety device.

THE OPEN-SIDE MACHINE FRAME

Another case of a design error hallowed by long and honorable engineering associations is the open-side or G-frame for punches and similar machines. Thousands of innocent engineering students have been taught to design such a frame as a straight beam subject to combined bending and tension. Professors of mathematics have grown gray-headed and round-shouldered trying to figure out how a stress which can exist only along a straight line, can also bend around a half-circle. The answer is that it doesn't and that the G-frame is a mechanical monstrosity.

Were a civil engineer to design a cast-iron or cast-steel structure to act in the same manner as a G-frame, he would not build a structure like Fig. 2, but one like Fig. 3. In Fig. 3 metal is provided to take care of the stresses that actually occur in the structure, the tensions in the flanges being gradually distributed as shears to the web; while in Fig. 2 the stresses in the flange, and between the flange and web, are greatly intensified where the flange is curved.

In setting forth some "horrible examples" the author has not had in mind the idea that these are the particular engineering and design errors which most require correction. Rather he wishes to point out the fact that a great many people who are unable to think intelligently about engineering problems are continually influencing engineering work, as customers for and users of engineering products, and that in the main this influence is very unfortunate. Also he has endeavored to point out that engineers themselves sometimes fall into slipshod methods in their work, accepting without question timeworn errors.

High-Speed Induction Motors and Frequency Changers

With Special Reference to Their Application to Woodworking, Grinding, Drilling, and Similar Machines Where High Speed Is of Special Advantage

By CHARLES FAIR,[1] WINCHENDON, MASS.

TO INCREASE production, to improve the quality of the product, and to reduce cost in a rising labor and material market is becoming increasingly difficult and dependent largely upon increasing the productiveness of machines.

It is the purpose of this paper to deal in a practical way with the general subject of induction frequency changers and high-speed motors, with special reference to their application to woodworking, grinding, drilling, and similar machines where high speed is advantageous, and to give sufficient data from which it is hoped the manufacturer will grasp the fundamentals sufficiently to apply them to his own problems.

The productive advantages of the direct-motor-driven machine are so well known that it will be unnecessary to dwell upon them here.

Motor speeds higher than those obtainable from the ordinary commercial frequencies (25 to 60 cycles) are now being extensively used in connection with such machines as shapers, molders, tenoners, carving, grinding, and drilling machines, etc. Inasmuch as the utmost in both mechanical and electrical refinement is now required of the direct-motor-driven machine, it is highly desirable to know how to obtain commercially the required higher motor speeds.

Since the squirrel-cage alternating-current motor is the ideal motor for high-speed-motor applications, and since three-phase current has been universally adopted, in order to avoid confusion this type of motor only will be considered.

The speed of a polyphase squirrel-cage type of induction motor is definitely fixed by its number of poles and the supply frequency. This can be expressed by the formula:

$$\text{R.p.m.} = \frac{60\,f}{\frac{N}{2}} = \frac{120\,f}{N}$$

where f is the frequency in cycles per second, and N the number of poles.

Fig. 1 shows the above formula plotted for various pole and frequency combinations. The highest speed obtainable from a given frequency is that of the two-pole motor. The synchronous speed (full-load speed slightly less) of a two-pole motor, stated in its simplest terms, is equal to sixty times the frequency, and when connected to a 60-cycle source of current it will run at 3600 r.p.m., on 50 cycles at 3000 r.p.m., on 40 cycles at 2400 r.p.m., on 30 cycles at 1800 r.p.m., and on 25 cycles at 1500 r.p.m. The motor in each case must have the proper winding. The above being the usual commercial frequencies, it is evident that if we are to obtain higher speeds than those mentioned we shall have to have higher frequencies.

The high-speed limit of the motor is mechanical, not electrical. Motors have been built commercially to operate on 420-cycle circuits, driving grinder spindles at 25,200 r.p.m. The rotors of the high-speed motors are of such simple and rugged construction that they will run at a peripheral speed as high as 14,000 ft. per min. with an ample factor of safety. For convenience of application the motors are made in several types, as the "drawn shell" with its smooth-ground stator, and the standard induction motor, frame type, with and without feet. Sizes vary from a fraction of a horsepower to 60 hp. or larger, and speeds up to 30,000 r.p.m. can be obtained, depending on rotor diameters.

[1] Chief Engineer, Baxter D. Whitney & Son, Inc. Mem. A.S.M.E.

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Frequencies higher than those of the usual commercial circuits can be obtained in various ways; two only will be considered here. The alternating-current generator, which can be built to generate any frequency and voltage desired, is so well known as to require no special description. It is, however, a necessity when high-speed induction motors are to be used and when an alternating-current source of supply, three-phase or two-phase, is not available. A second method, and one that is becoming increasingly popular due to its simplicity, low first cost, and reliability, is that of boosting the frequency of existing alternating-current circuits by means of the induction-type frequency changer.

The induction-type frequency changer is essentially a slip-ring or wound-rotor-type induction motor, usually driven backward,

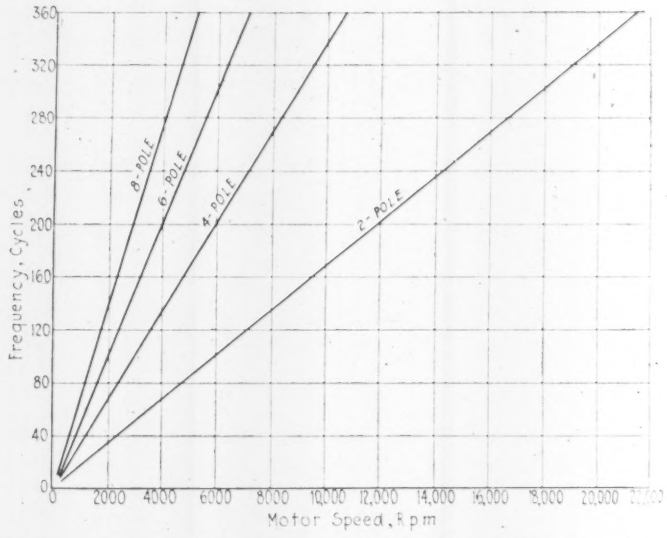

FIG. 1 INDUCTION-MOTOR SPEED CURVES FOR VARIOUS POLE AND FREQUENCY COMBINATIONS

that is, in the opposite direction from that which it would run as a motor. It can be used wherever alternating current, three-phase or two-phase, is available. The primary or available alternating-current source of supply is usually connected to the stator of the frequency changer, and the high-frequency or secondary source of supply is obtained at the slip rings. No regulator is required.

FREQUENCY CHANGERS FOR BOOSTING FREQUENCIES

The frequency changers considered here are those used for boosting frequencies, and may for convenience be divided into two general groups.

In the first group are frequency changers used to boost the regular commercial standard frequencies and voltages in the proper ratio, so that standard motors already in use can be speeded up. For example, a two-pole, 60-cycle, 220-volt, 3600-r.p.m. motor could be connected to an 80-cycle, 293-volt circuit, and would run at 4800 r.p.m.; or again a 30-cycle, 220-volt, 2-pole, 1800-r.p.m. motor when connected to a 60-cycle, 440-volt circuit would run at 3600 r.p.m. Within this group it is possible to speed up many standard motors provided the voltage is in proper relationship to the frequency and not too high for the insulation or accepted shop practice. It is seldom that motors in this group are run beyond 120-cycle speeds. When speeding up standard motors, mechanical features such as rotor stresses due to speed, bearings, and the tendency to draw oil from the bearing reservoir would necessarily have to be considered. Unless the characteristics of the motors in

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question are pretty well known it would be well to consult the motor manufacturer before speeding up the standard motors referred to in this group, especially motors of over 15 hp. or of more than two poles. By "standard" motors are meant motors built to operate on normal commercial frequencies. Within this group are included a great variety of motors of the regular shaftless type, as well as of the drawn-shell type.

The second group, on account of the wide range of frequencies covered, is divided into two series. In the first series are frequency changers used to boost any of the standard frequencies, 25 to 60 cycles, 220, 440, 550 volts, two-phase or three-phase, to a new but established standard with a normal rating of 120 cycles at 225 volts, irrespective of the primary voltage. In the second series frequency changers are used to boost any of the standard frequencies to a new standard with a normal rating of 200 cycles at 114 volts, irrespective of the primary voltage.

Very early in the development of the high-speed motors it was evident that standards would have to be established to avoid confusion and complications. Obviously it would only be inviting trouble if we attempted to run a 60-cycle, 440-volt motor on a 180-cycle, 1320-volt circuit, or if we attempted to operate a 60-cycle, 220-volt motor on a 420-cycle, 1540-volt circuit.

Frequency changers and the motors of the second group, first series, are designed to operate through a wide speed range varying from 80 cycles at 150 volts to 199 cycles at 373 volts, and correspond to a two-pole-motor synchronous speed of 4800 to 11,940 r.p.m. This wide range of frequencies and the corresponding motor speeds can be obtained in a single frequency changer and motor, provided the mechanical speed limit of the frequency changer or motor in question has not been exceeded. Because at 199 cycles in this series we have reached 373 volts, it was considered advisable to adopt a second series of standard frequency ratings with 200 cycles at 114 volts as normal, and to boost any of the standard commercial frequencies, three-phase or two-phase, to 200 cycles and upward. This second series of frequency changers and the motors to operate from them start out with a basic rating of 200 cycles at 114 volts, and will operate throughout a range with the proper voltage of from 200 cycles at 114 volts up to 500 cycles at 285 volts. The speed of a two-pole motor corresponding to this range of frequencies is from 12,000 to 30,000 r.p.m. Here again the speed limitation is mechanical, not electrical. The minimum motor speed is largely a matter of ventilation. Large frequency changers and frequency changers boosting at a relatively high frequency ratio would necessarily have a greater number of poles to prevent excessive speed.

The speed of the frequency changer at a given secondary frequency is fixed by the primary or excitation frequency and the number of poles of the frequency changer. The kilowatt rating is usually given at a definite secondary frequency and voltage, and will vary from this rating practically in proportion to the variation in the secondary frequency. The output of the two-pole motors in sizes generally used on the higher-frequency circuits will increase or decrease somewhat less than at a straight frequency ratio, depending upon how far the frequency varies from that at which the motors were rated.

Drive for Frequency Changers

Frequency changers can be driven from any source of power available, but the excitation must be alternating current, three-phase or two-phase. By rotating the machine in the opposite direction from that which it would normally run as a motor, the frequency generated by rotation is automatically added to the frequency of the excitation circuit. If the rotor be driven backward at normal speed, that is, the speed at which it would run as a motor on the primary frequency, then the normal line frequency will be obtained as through a transformer, plus an equal number of cycles due to rotation.

To increase or decrease the secondary frequency of a given machine it is only necessary to increase or decrease the speed of the frequency changer. The secondary frequency is equal to the frequency of excitation, which remains constant, plus the variable or generated frequency, which is a function of the speed of the rotor. The speed of a frequency changer can be expressed by the formula:

$$\text{R.p.m.} = 60\,\frac{(S-P)}{\frac{N}{2}} = 120\,\frac{(S-P)}{N}$$

where S = secondary frequency or cycles per second
P = primary frequency
N = number of poles.

To illustrate: At what speed must we drive a six-pole frequency changer excited from a 60-cycle source to obtain 150 cycles?

R.p.m. = 120(150 — 60)/6 = 1800, which can be read from Fig. 2 by subtracting the primary from the secondary frequency.

A direct-connected squirrel-cage motor is the ideal drive for frequency changers when the speed of the latter is such that a direct-connected motor can be used. Belt-driven frequency changers can be run to obtain any desired secondary frequency within the machine's range. It is sometimes desirable for manufacturing reasons to obtain varying frequencies from the same

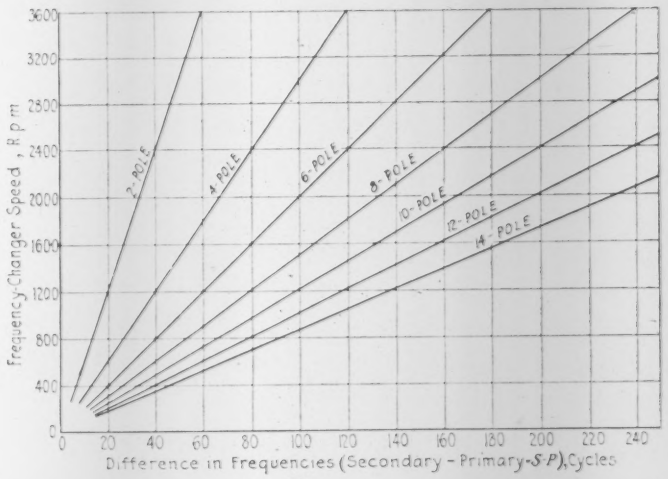

FIG. 2 CURVES TO DETERMINE SPEED AT WHICH FREQUENCY CHANGER MUST RUN FOR VARIOUS POLE AND FREQUENCY COMBINATIONS

frequency changer, in which case the direct-connected a.c. or d.c. adjustable-speed motor, or the a.c. multi-speed, can be used advantageously.

It is to be understood, however, that when the speed of a frequency changer is changed the speed of all motors connected to its secondary will also be changed.

Frequency changers are usually rated in kilowatts, 0.8 power factor, and at a definite secondary frequency; but due to the fact that they can be run at such a wide range of speeds, approximately three to one, the actual kilowatts will be greater or less than the rated kilowatts, depending upon the speed at which the frequency changer is driven. If the kilowatt rating is given at 120 cycles and the machine is run at a speed to give 180 cycles secondary, then the kilowatt output would be increased 50 per cent. Likewise, if the same machine were run to deliver 90 cycles, the output would be reduced 25 per cent.

Power Required to Drive Frequency Changers

There has always been a great deal of confusion over the question of the power required to drive frequency changers at various speeds, due, no doubt, to an imperfect understanding of the workings of the machine. To facilitate this the following formulas which take into account efficiency have been developed.

Let P be the number of primary cycles per second, S the number of secondary cycles, F the frequency at which the kilowatt rating is given, and R the kilowatt rating at frequency F. Then

$$\text{Hp. to drive} = R\left(\frac{1000}{746 \times 0.9}\right)\left(\frac{S-P}{S}\right)\left(\frac{S}{F}\right)$$
$$= R\left(1.4894\right)\left(\frac{S-P}{F}\right)$$

EXAMPLE:—Required, the horsepower to drive a 70-kw. frequency changer at 110 cycles when the frequency changer is rated at 100 cycles and

the primary is 50 cycles. By substitution of these values in the formula, the power to drive will be found to be 62.5 hp.

The above general formula simplified for two special cases would be as follows:

Case I. When $F = S$ or the kilowatt rating is given at the operating secondary frequency, then

$$\text{Hp. to drive} = R(1.4894)\,\frac{(S - P)}{S}$$

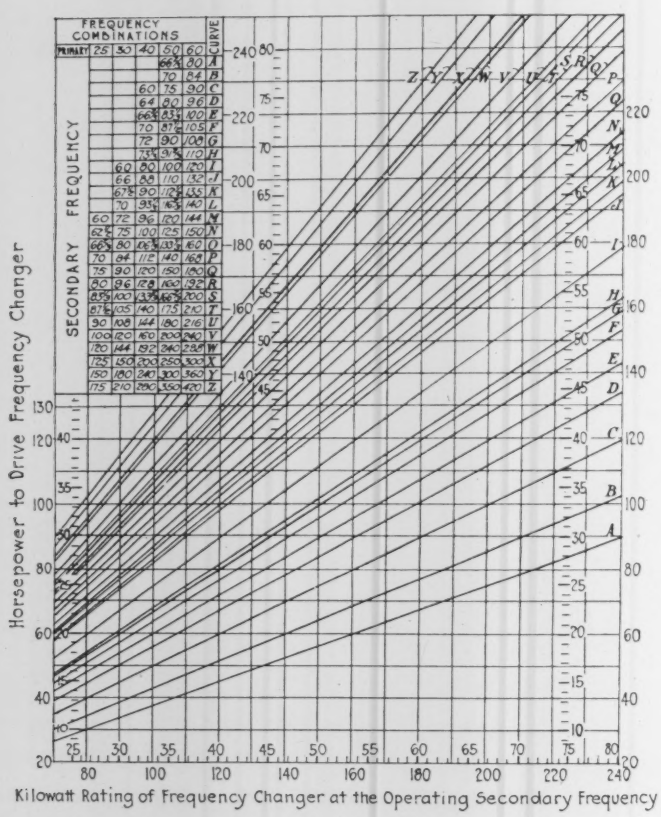
FREQUENCY COMBINATIONS
SECONDARY FREQUENCY
Horsepower to Drive Frequency Changer
Kilowatt Rating of Frequency Changer at the Operating Secondary Frequency

FIG. 3 HORSEPOWER TO DRIVE FREQUENCY CHANGERS AT VARIOUS FREQUENCY COMBINATIONS WHEN THE KILOWATT RATING IS GIVEN AT THE OPERATING SECONDARY FREQUENCY

EXAMPLE:—Required, the horsepower to drive a 70-kw. frequency changer rated and to be run at 100 cycles secondary; primary, 50 cycles. By substitution, hp. = 52.13.

Case II. When $F = S = 120$ cycles, the frequency at which the machine is to run, then

$$\text{Hp.} = R\left(1.4894\right)\left(\frac{S - P}{120}\right) = R\,(0.01241)\,(S - P)$$

EXAMPLE:—Required, the horsepower to drive a 70-kw. frequency changer, secondary 120 cycles, primary 40 cycles. By substitution, hp. = 69.5.

The formulas for Cases 1 and 2 have been reduced to curves that can be read directly from Fig. 3 and Fig. 4, respectively.

The general formula can be read from the kilowatt-rating curve, Fig. 5, and then directly from Fig. 3.

EXPLANATION OF CURVES

Five curves have been prepared to simplify calculations in connection with determining the proper frequency changer and motor speeds and the horsepower to drive. The first two curves, Figs. 1 and 2, are for speeds of motors and frequency changers. The last three curves, Figs. 3, 4, and 5, are for determining the horsepower necessary to drive a given frequency changer at any frequency combination.

MOTOR AND FREQUENCY-CHANGER SPEEDS

Fig. 1 gives induction-motor speed curves for various pole and frequency combinations. Read from "frequency, cycles" at the left of figure across to curve for number of poles, and then down to obtain motor speed. If the speed and number of poles are given, read from bottom line "motor speeds" up to curve for the number of poles and across to "frequency, cycles" to obtain frequency at which motors must run.

Fig. 2 is a curve for use in determining the speed at which the frequency changer must run for various pole and frequency combinations. In employing it, first subtract the primary or excitation frequency from the secondary or new frequency at which the motors are to run. The speed of the frequency changer can then be obtained by running from the value just found on the line "difference in frequencies" to the curve for the number of poles of the frequency changer, and then across to line "frequency-changer speed." To obtain the secondary frequency when the motor speed is known, consult Fig. 1.

HORSEPOWER REQUIRED TO DRIVE FREQUENCY CHANGERS

Three curves, Figs. 3, 4, and 5, were prepared primarily to determine the amount of mechanical power required to drive frequency changers. There are three general conditions to which the curves apply: first, when the kilowatt rating is given at the operating secondary frequency at which the machine is to be run; second, when the rating is given at 120 cycles regardless of the secondary frequency at which the machine is to run; and third, when it is given at neither 120 cycles nor the operating secondary frequency at which it is to run.

Fig. 3 applies to the first condition, Case I, and its use can be explained best by an example. Required, the horsepower to drive a

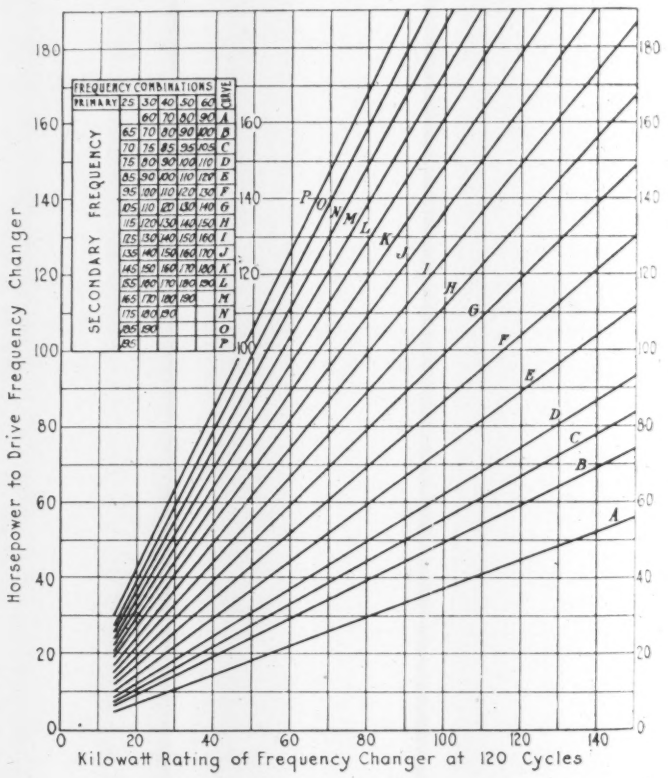
FREQUENCY COMBINATIONS
SECONDARY FREQUENCY
Horsepower to Drive Frequency Changer
Kilowatt Rating of Frequency Changer at 120 Cycles

FIG. 4 HORSEPOWER TO DRIVE FREQUENCY CHANGER AT VARIOUS FREQUENCY COMBINATIONS WHEN THE KILOWATT RATING IS GIVEN AT THE SECONDARY FREQUENCY OF 120 CYCLES

frequency changer with 60 cycles primary and 150 cycles secondary, and rated 40 kw. at 150 cycles. From the table in the upper left-hand corner of the chart, follow down the 60-cycle primary column to 150 cycles secondary, opposite which will be found the letter "N" designating the curve to be used. At the bottom of the chart will be found numbers corresponding to the kilowatt ratings. Follow up the 40-kw. line until it intersects curve N. From this intersection follow over to the left side of chart and read the horsepower to drive, which in this case is 35.75.

It should be noted that there are two scales on Fig. 3; a coarse scale for sizes from 25 to 80 kw., and a finer one for 80 to 240 kw.

For sizes smaller than 25 kw. it is only necessary to read values ten times and move the decimal point. The coarse scale for kilowatt rating corresponds to the coarse scale for horsepower to drive, and the fine scale for kilowatt rating to the fine scale for horsepower to drive.

Fig. 4 applies to the second condition, Case II, when the kilowatt rating is given at 120 cycles but the machine is to run at some other frequency. Required, the horsepower to drive a 40-kw. machine, 60 cycles primary rated at 120 cycles, but to be run at 150 cycles. Referring to the table, Fig. 4, the curve for a 60/150 cycle combination will be found to be designated as "H." Following up the 40-kw. line to curve H, the power to drive will be found to be 44.67 hp.

Figs. 4 and 5 must be used for the third condition. Assume a frequency changer rated at 50 kw. at 90 cycles and boosting from 50 to 100 cycles. Required, the horsepower to drive. Referring to Fig. 5, numbers will be found at the left which designate the rating in kilowatts. Follow the 50-kw. line from the left until it intersects the 90-cycle line. From this point follow up to the 120-cycle line and read what the basic kilowatt rating at 120 cycles would be—66.67 in this case. Using this new rating, find the horsepower to drive as before from Fig. 4. The 50/100 curve is "D" and the horsepower to drive, 41.37.

Fig. 5 can also be used to determine the actual kilowatt output

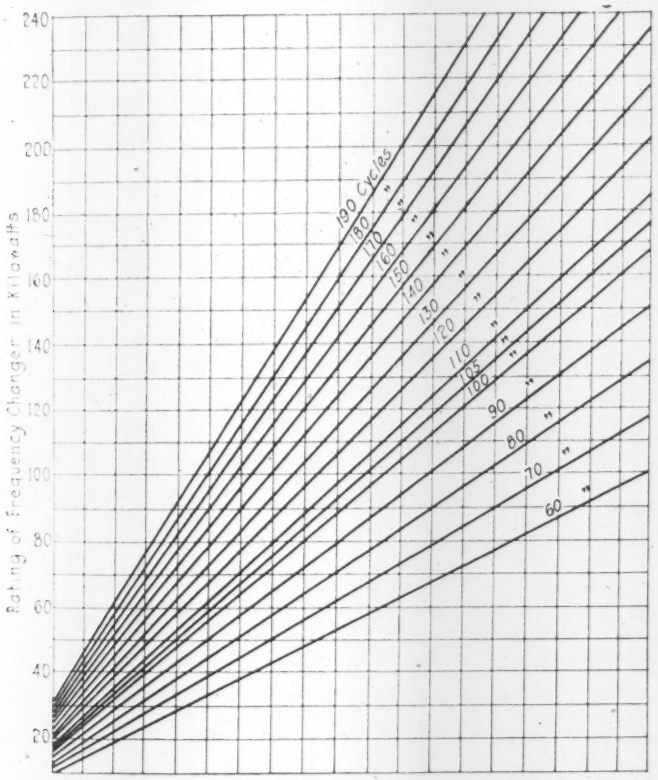

FIG. 5 CURVE SHOWING RELATION BETWEEN ACTUAL KILOWATT OUTPUT OF FREQUENCY CHANGER OPERATING AT VARIOUS FREQUENCIES AND ITS BASIC RATING

of a frequency changer at any frequency if its rating is known at some given frequency. Read from the given rating at the left side of chart to the frequency curve at which rating was given, and then up or down to the frequency curve at which the machine is to run, thence back to the kilowatt rating at the left.

The excitation of a frequency changer at full load is to the total energy as the primary frequency is to the secondary frequency. Similarly the mechanical energy to rotate the machine is to the total energy as the difference of frequencies is to the secondary frequency.

$$\frac{\text{Electrical}}{\text{Total}} = \frac{P}{S} \qquad \frac{\text{Mechanical}}{\text{Total}} = \frac{S - P}{S}$$

When boosting from one of the lower standard frequencies, 25–60 cycles, to a higher frequency, the secondary should be three-phase and the voltage should correspond to one of the established standards, as 110 volts or 220 volts at 60 cycles with corresponding frequency and voltage up to 199 cycles, or the high-voltage limitation when starting with 220 volts at 60 cycles. The second series begins with 200 cycles, 110 volts, and runs up to 500 cycles, 275 volts and beyond. Standard motors can then be used and interchangeability maintained to a large extent.

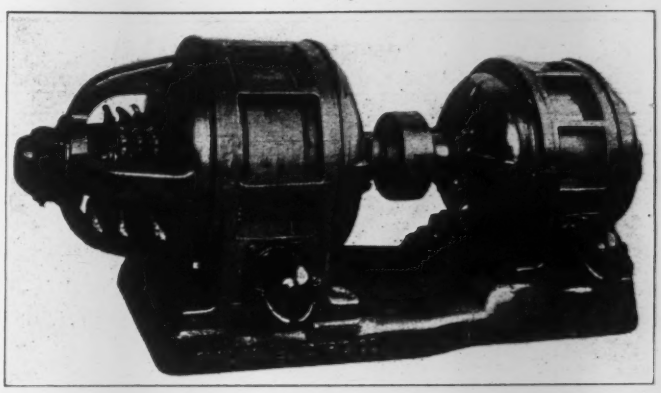

FIG. 6 INDUCTION FREQUENCY CHANGER DRIVEN BY A DIRECT-CONNECTED INDUCTION MOTOR

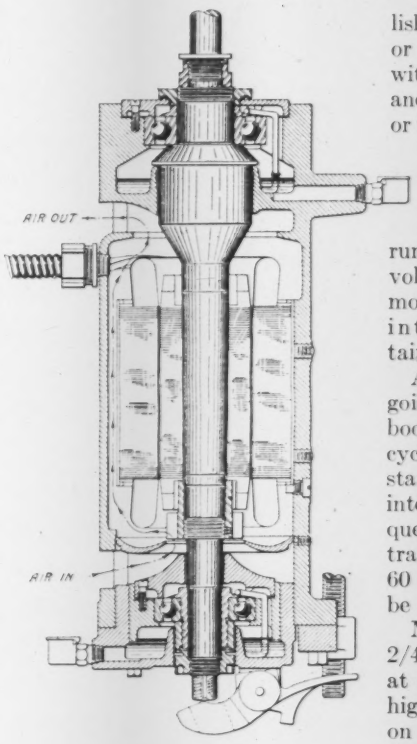

FIG. 7 SECTIONAL VIEW OF A BALL-BEARING 15-HP. 5400-R.P.M. VERTICAL SHAPER SPINDLE

An exception to the foregoing might be the case of boosting from 25, 30, or 40 cycles to only 60 cycles as a standard; and when it is not intended to boost the frequency any higher, and where transmission is a factor, then 60 cycles at 440 volts would be considered.

Motors can be wound for 2/4 pole combinations to run at speed or half-speed on the higher frequencies as well as on standard frequencies. When the voltage is in the proper ratio to the frequency the same motor can be run on two or more frequencies as, for instance, on a 110-volt, 60-cycle, or 220-volt, 120-cycle circuit.

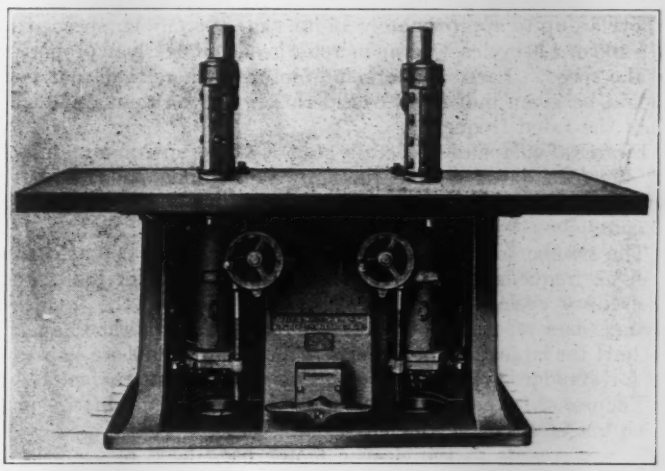

FIG. 8 WHITNEY HEAVY-DUTY DOUBLE BALL-BEARING SHAPER WITH ROTORS OF 15-HP. MOTORS MOUNTED DIRECTLY ON THE SHAPER SPINDLES

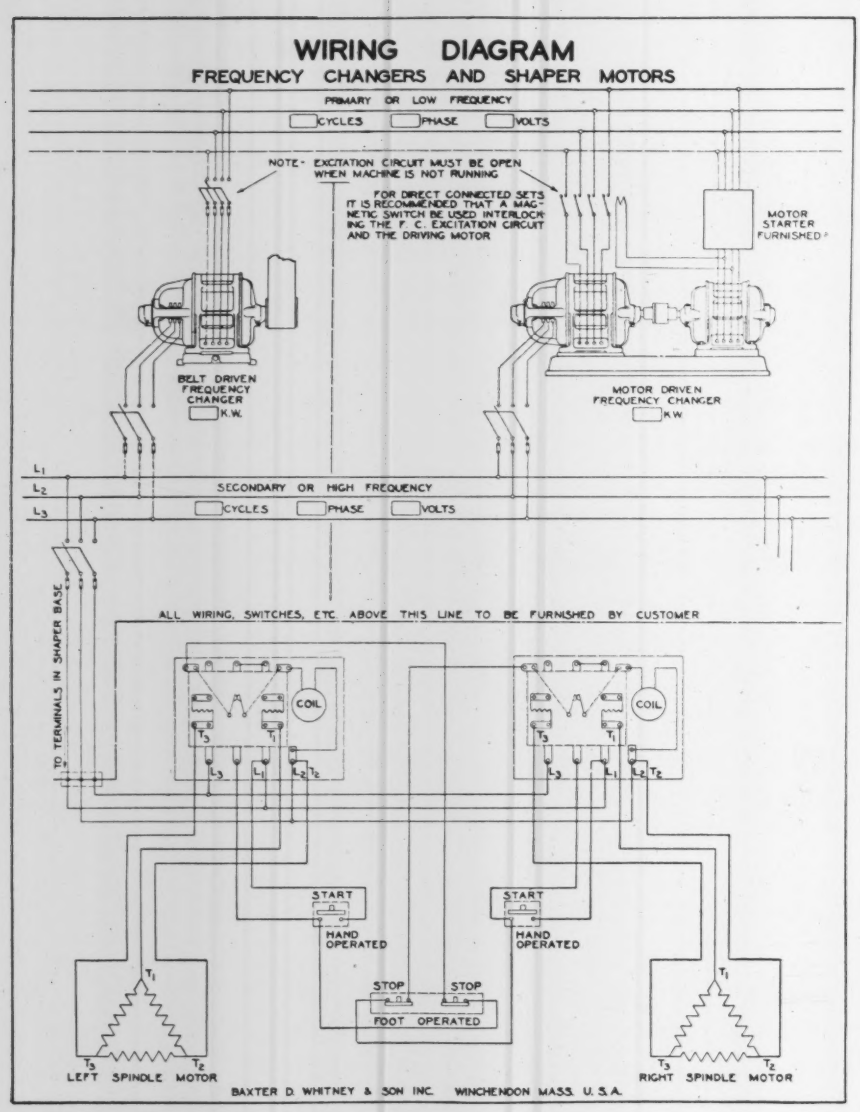
WIRING DIAGRAM
FREQUENCY CHANGERS AND SHAPER MOTORS
PRIMARY OR LOW FREQUENCY
CYCLES PHASE VOLTS
NOTE- EXCITATION CIRCUIT MUST BE OPEN WHEN MACHINE IS NOT RUNNING
FOR DIRECT CONNECTED SETS IT IS RECOMMENDED THAT A MAGNETIC SWITCH BE USED INTERLOCKING THE F. C. EXCITATION CIRCUIT AND THE DRIVING MOTOR
MOTOR STARTER FURNISHED
BELT DRIVEN FREQUENCY CHANGER K.W.
MOTOR DRIVEN FREQUENCY CHANGER K.W.
L1
L2
L3
SECONDARY OR HIGH FREQUENCY
CYCLES PHASE VOLTS
ALL WIRING, SWITCHES, ETC. ABOVE THIS LINE TO BE FURNISHED BY CUSTOMER
TO TERMINALS IN SHAPER BASE
COIL
COIL
START
HAND OPERATED
START
HAND OPERATED
STOP
STOP
FOOT OPERATED
LEFT SPINDLE MOTOR
RIGHT SPINDLE MOTOR
BAXTER D. WHITNEY & SON INC. WINCHENDON MASS. U.S.A.

Fig. 9 Wiring Diagram for a Two-Spindle Motor-Driven Shaper with Hand-Operated Push-Button Start Stations and Foot-Operated Safety-Stop Stations

The output of two-pole motors will increase or decrease from normal rating approximately in proportion to the increase or decrease in speed up to 120 cycles and beyond, heating being the limiting factor at slow speeds. Four-pole motors, 110 and 220 volts at 60 cycles, up to approximately 15 hp. can generally be speeded up to a 100 or 120 cycles, with an increase in output nearly in proportion to the speed. Larger motors and motors of more than four poles should be given individual consideration before running at higher than the rated frequency.

Electrical companies generally do not advise speeding up motors of standard bearing construction. It is understood that the bearings of high-speed-motor applications are a part of the machine design.

The number of motors that can be successfully operated from a given frequency changer depends upon the size of the motors, their duty cycle, and the power factor. Because the frequency changer is a separate power unit, it must be of sufficient capacity to start the largest motor; otherwise the requirements are estimated as for standard frequencies, with due regard to power factor.

The use of frequency changers for boosting standard frequencies is no longer either new or novel.

Improvements in the electric motor have been responsible for marked changes in the design of machine tools.

Woodworking-machinery manufacturers alone are using thousands of high-speed and built-in motors. The percentage of motor-driven to belt-driven machines has materially increased in the last few years, one company having developed the direct-motor-driven machine to the point where its production of this type has increased in the past six years from 30 per cent to 90 per cent of their output, making the belt-driven machine seem almost obsolete.

Frequency Changers and High-Speed-Motor Applications

It has been said that manufacturers are necessarily slow in commercializing scientific inventions. Fortunately there are instances which have proven exceptions to the rule. The manufacturer who fails to recognize changing conditions necessarily falls behind.

Since it is recognized that insufficient speed or a lack of power in a machine is a source of expense and because of the known difficulties connected with high-speed mechanical transmissions, the direct-connected motor is ideal where speed is a factor.

The successful machine designer of today must have a working knowledge of electric motors in order to apply them advantageously to his machine.

High-speed motors are frequently designed to the smallest physical dimensions, and, on account of compactness, the matter of ventilation becomes a most important matter. Rotors are often small in diameter in proportion to their output, and for that reason relatively large shafts or spindles are used. Large shafts, collars, nuts, etc. further complicate the ventilating problem to the extent that an auxiliary fan is often a necessity. One of the approved methods for ventilating a high-speed motor by the use of an auxiliary fan is clearly shown in Fig. 7, which is a sectional view of a ball-bearing 15-hp. 5400-r.p.m. vertical shaper spindle. The complete machine is shown in Fig. 8. The oil reservoir and method of lubricating the bearing which prevents the oil from getting into the motor windings, should be noted. The projecting spindle with its foot-operated lock to hold the spindle, and which is automatically released when the foot is removed, is one of the newer mechanical developments that prevent the motor from being burnt out due to a locked rotor.

As the speed of the motor is increased, the tendency to draw oil from the bearings becomes greater due to increased windage and suction. Bearings should be designed to prevent oil seepage and not to unduly wear and allow the rotor to touch the stator as the radial air gap of the motor is not large.

Owing to the compactness of the modern built-in motor-driven machine with its multiplicity of motors and the necessity for protecting both the workman and the machine, it often becomes necessary for the manufacturer to wire the machine ready to connect to the power lines. Fig. 9 shows one of the methods of connecting a two-spindle motor-driven shaper with hand-operated push-button start stations and foot-operated stop stations. Motor under-voltage protection is a necessity in a case of this kind to protect

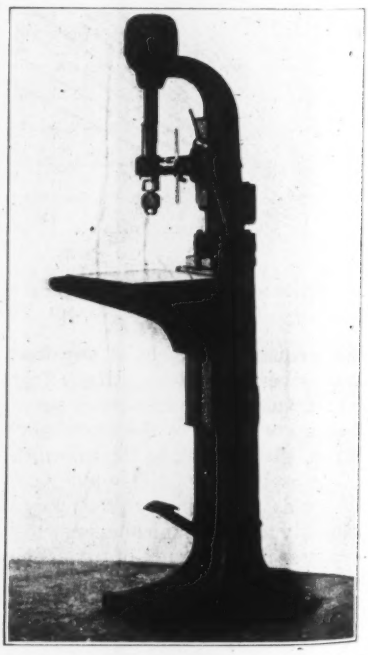

Fig. 10 High-Speed Drill, 2/4-Pole 180-Cycle Motor, 10,800 and 5400 R.P.M.

the operator against the machine's starting up unexpectedly.

The tabulation below shows the speed combinations that can be obtained with a two-pole motor when connected to the secondary of a four-pole or a six-pole frequency changer driven by a multi-speed 600/900/1200/1800-r.p.m. a.c. motor, the primary or excitation source being 60 cycles.

Speed of driving motor, r.p.m.	4-Pole Frequency Changer — Secondary cycles	4-Pole Frequency Changer — Two-pole motor speed, r.p.m.	6-Pole Frequency Changer — Secondary cycles	6-Pole Frequency Changer — Two-pole motor speed, r.p.m.
1800	120	7200	150	9000
1200	100	6000	120	7200
900	90	5400	105	6300
600	80	4800	90	5400

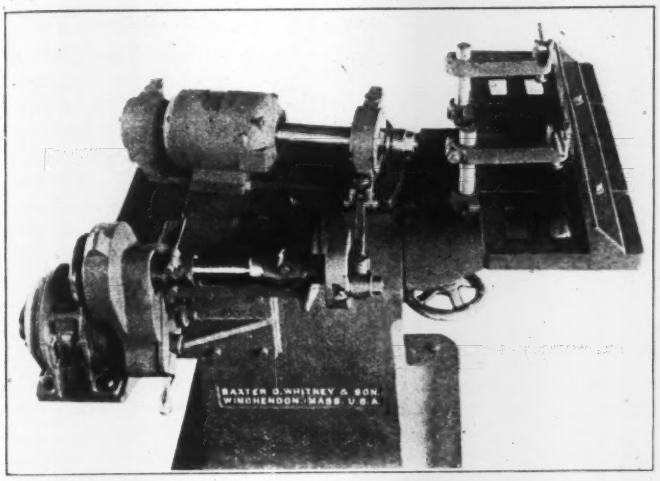

FIG. 11 VIBRATING-BIT MORTISER, 120-CYCLE, 7200 R.P.M., DIRECT-MOTOR-DRIVEN SPINDLE AND 60-CYCLE FEED MOTOR

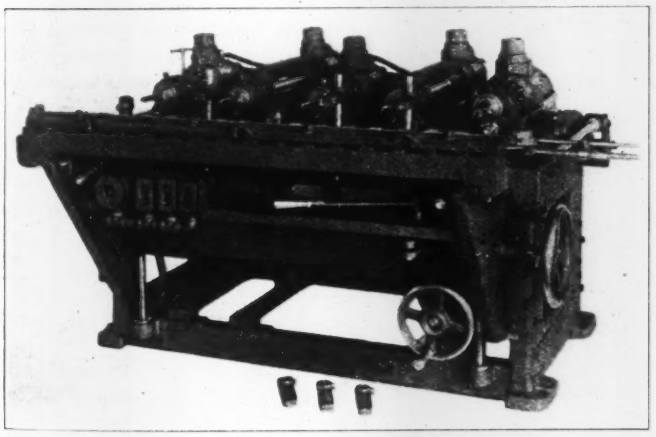

FIG. 12 MULTIPLE MORTISER WITH FIVE HIGH-SPEED BUILT-IN MOTORS AND ONE FEED MOTOR

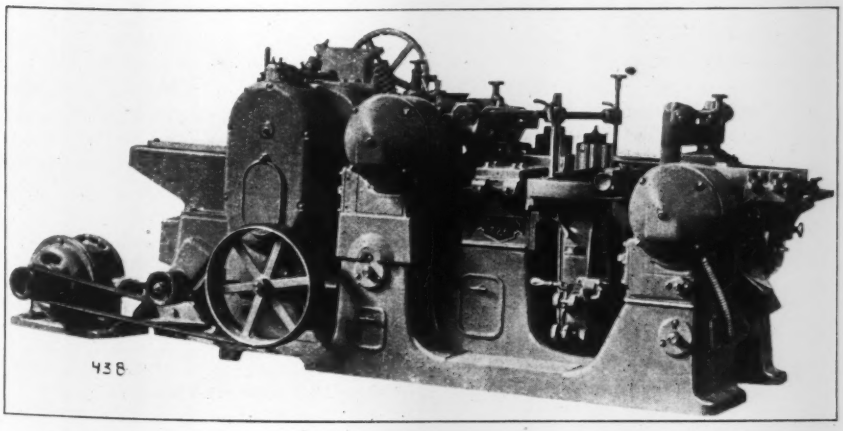

FIG. 15 REAR VIEW OF MOLDER, SHOWING FOUR BUILT-IN 90-CYCLE MOTORS, 5400 R.P.M., AND ONE FOUR-SPEED 60-CYCLE FEED MOTOR

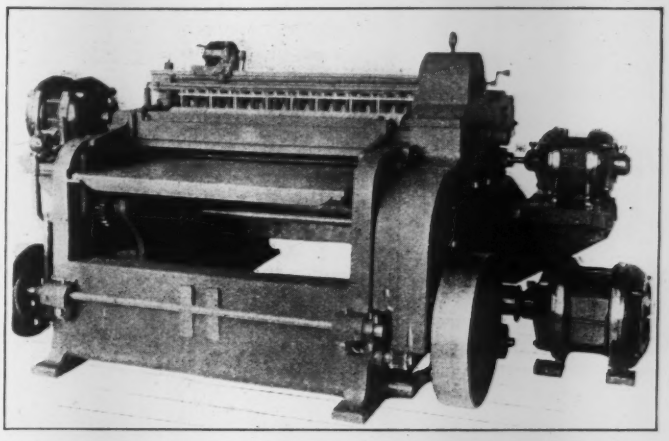

FIG. 13 DOUBLE PLANER, ENDLESS-BELT FEED, TWO 4-POLE MOTORS OPERATING ON 120 CYCLES AT 3600 R.P.M. AND ONE 6-POLE 1800-R.P.M. MOTOR OPERATING ON A 40-CYCLE CIRCUIT

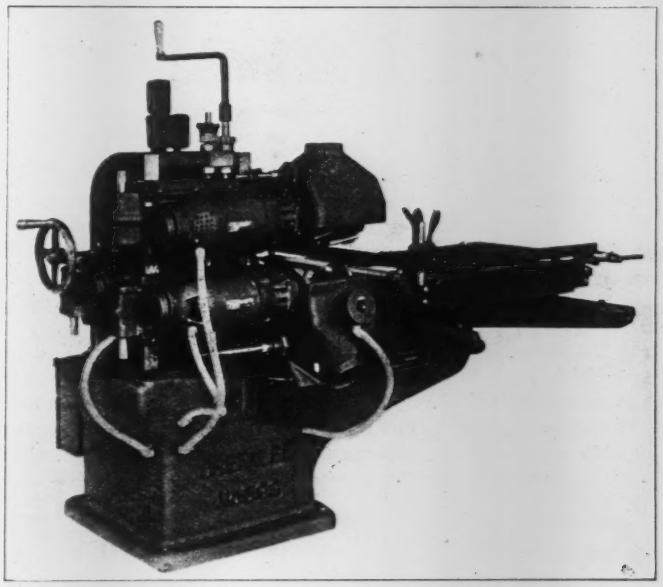

FIG. 14 SINGLE-END TENONER WITH THREE HIGH-SPEED BUILT-IN-TYPE MOTORS

It is sometimes desirable to run several circuits of different frequencies and at the same time be able to change the frequency of the circuits at will. This can be accomplished, within the range of the set by driving the frequency changer with either a d.c. or an a.c. adjustable-speed motor. When multiple circuits of this kind are used the machines are usually wired so that they can be plugged in on any one of the various circuits. The advantage of a flexible arrangement of this kind is evident.

The Leland-Gifford high-speed drill shown in Fig. 10 is a good example of compact and clean-cut construction. The motor is a 2/4-pole type, and in this case operates on a 180-cycle circuit giving speeds of 10,800 and 5400 r.p.m. The motor-starting switch is conveniently located on the right side of the machine and the pole-changing switch on the left side, the mechanism of both switches being entirely enclosed within the goose neck.

Fig. 11 shows a vibrating-bit mortiser. The spindle is driven by a direct-connected 4-hp. 120-cycle motor running at 7200 r.p.m. and the feed by a $^3/_4$-hp., 60-cycle motor. The head block and spindle swing through a 6-in. stroke at the rate of 150 or 200 strokes per minute. No belts or belt tighteners are required to take care of the vibrating spindle.

The Greenlee multiple mortiser shown in Fig. 12

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is another example of a compact motor-driven high-speed machine. Five drawn shell-type motors are built into the heads, and one standard motor drives the feed.

The Whitney double planer shown in Fig. 13 is driven by two four-pole motors operating on a 120-cycle circuit at 3600 r.p.m. and one six-pole 800-r.p.m. motor operating on a 40-cycle circuit. From the 120-cycle circuit motors can be run two-pole at 7200 r.p.m. and four-pole at 3600 r.p.m., while the lower speeds can be obtained from the 40-cycle circuit. In this case a 40-cycle shop circuit only was available, and it was necessary to boost to 120 cycles on account of running shapers and other machines at 7200 r.p.m. Speeds of 2400 r.p.m. and slower would be obtained by using 40-cycle motors.

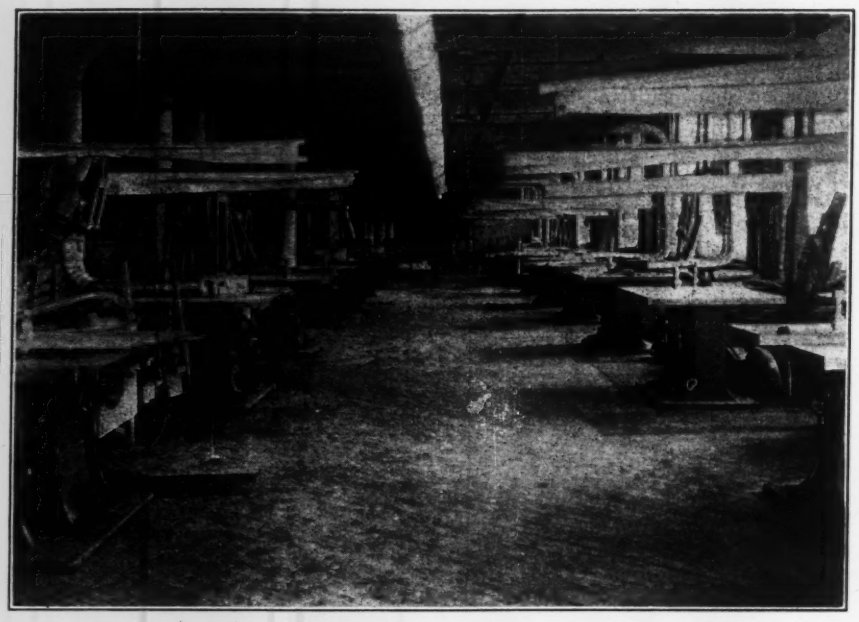

Fig. 16 Battery of Motor-Driven Shapers Installed in an Automobile Body Plant and Operating on a 105-Cycle Circuit, 6300 R.P.M.

The Mattison molder, Fig. 14, is an excellent example of a compact high-production motor-driven machine. Four 90-cycle motors running at 5400 r.p.m. are direct-connected to the cutter heads, while the feed is driven by a 60-cycle multi-speed motor running 600/900/1200/1800 r.p.m.

Fig. 16 shows one of a number of similar batteries of Whitney shapers installed in automobile body plants. The machine spindles are all direct-motor-driven, and in this particular instance run from a 105-cycle circuit at 6300 r.p.m. Production is so rapid on this type of machine that it is impossible to confine cuttings and chips within the bounds of a practical exhaust system; therefore the drop curtains to protect the operator.

Many manufacturers are still worrying about the power cost of direct-connected motor-driven machines and want to know whether the installation of many motors will not increase their power bills. The answer depends somewhat on their present equipment, but there is no question but that a machine with a reasonably efficient drive will take more power if driven to greater capacity. The real point to consider is obviously the relationship of power cost to productive cost. The cost of power in the average manufacturing plant, exclusive of plants where power is a main item of productive cost—as in the manufacturing of graphite and aluminum—is roughly from one to three per cent of the productive cost, while labor is roughly 50 per cent. If by speeding up machines or by installing more productive machines the production could be increased 15 per cent, then $7\frac{1}{2}$ per cent of the saving would be represented by labor, while the power cost might have increased 15 per cent or even more. If $1\frac{1}{2}$ per cent represented the original cost, then a 15 per cent increase in power cost would represent only 0.225 per cent increase in productive cost. The saving would therefore be more than 7 per cent. Conclusions drawn from the above are obvious.

Graphical Methods of Calculation

The Construction of Cartesian Diagrams and Alignment Charts for Use in the Rapid Determination of the Values of Unknown Quantities in Mathematical Expressions

By HERBERT L. SEWARD,[1] NEW HAVEN, CONN.

ALL graphical methods might be divided into two classes: (*a*), where the length of a line is important, as in vector analysis; and (*b*), where the position of a line or point is important but the length of the line is unimportant.

Graphical methods are used in general in connection with algebraic quantities:

1 To determine a law or formula to fit observations
2 To determine the value of the unknowns in a given mathematical expression.

This paper will deal with the latter case and will make use of method (*b*) mentioned above. The French have long called this branch of mathematics Nomographie, "the writing of a law," but that word has made little progress in this country. The French nomogram or abaque is variously styled in this country as a diagram, chart, or graphical solution.

The construction and use of a permanent diagram for a given mathematical expression can be made to depend on the following two methods:

I Where three straight lines intersect at a point (Cartesian diagram)
II Where three points lie on a straight line (alignment chart). See Fig. 1.

In constructing such diagrams it is not necessary to make numerous tedious solutions of the formula, tabulate the results, and expect to arrive at a useful diagram. The method is rather to select from the formula certain functions, and by strategically spacing certain scales in the proper position the points or lines which serve to give all solutions of the formula may then be neatly constructed.

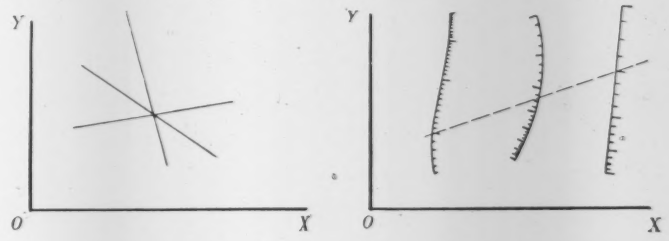

FIG. 1 CARTESIAN DIAGRAM AND ALIGNMENT CHART

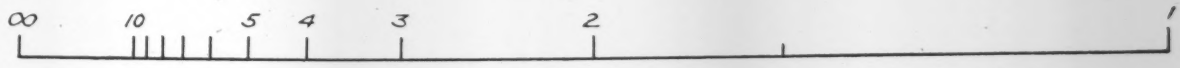

FIG. 2 FUNCTION SCALE FOR $y = 1/x$

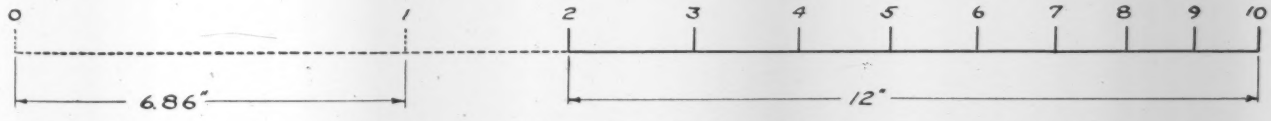

FIG. 3 PLOT OF SQUARE-ROOT SCALE

The slide rule and logarithmic cross-section paper are now very familiar devices and are examples of the use of *function scales*. We should enlarge our ideas of such scales to include any function of a single variable as the basis of a straight-line scale and of any two variables as the basis of a binary scale, to be defined later. These scales are to be laid down in such a way that the points or lines derived from them constitute a neat picture of the whole range of use of the given formula.

The function scale may be defined as a series of values of the function measured off on a straight line but inscribed with the values of the variable. Given, $y = f(x)$, the values of y are measured off, but after measurements are made, each segment is inscribed with the corresponding value of x. Fig. 2 shows a function scale for $y = 1/x$.

Each variable in a given formula will cover a certain useful range, and it is desirable to construct the scales to cover only that range. The *scale factor* may be chosen so as to accommodate the desired range and still make a scale any desired overall length. The relation between L, the length of the scale in inches, M, the scale factor, x_1, the upper limit of the variable, and x_2, the lower limit of the variable, is

$$L = M[f(x_1) - f(x_2)]$$

from which M may be computed. For example, if it is desired to plot a square-root scale between the limits $x_1 = 10$ and $x_2 = 2$ on a line twelve inches long, the modulus M would equal twelve inches divided by the difference between the square root of 10 and the square root of 2, or 6.86 inches. With this unit of measurement the scale will appear as in Fig. 3.

GENERAL THEORY OF REPRESENTING EQUATIONS OF THREE VARIABLES—CARTESIAN DIAGRAMS

The general theory of representing equations of three variables will first be considered, because equations of more than three variables are readily handled by making use of extensions of this theory.

Cartesian diagrams may be considered as falling into three general classes:

1 Radial straight lines
2 Parallel straight lines
3 General linear, tangent to envelope.

In general it may be said that the ability to algebraically arrange a given formula in a form similar to the various type forms is more important and takes more ingenuity than merely following the formal procedure of each of the standard forms.

Omitting the general case where a family of curves may be plotted upon coördinate paper using regular scales upon the axes with possibly different scale factors on the two axes, as shown in Fig. 4, we shall consider the example of the radial-type straight-line diagram. We shall prefer to draw straight lines rather than curves—but if curves must be drawn, it is well to select that variable which appears only in integral numbers as the variable to be represented by the family of curves. Many times the plotting of such a family of curves may be expedited by observing that the curves cut regular scales upon some system of lines. In Fig. 4 the portion of each ordinate between the extreme curves is cut into equal segments by the curves.

If an equation may be thrown into the form $y = mx$, the resulting diagram will be of the "radial" type. The equation of Fig. 4, $S = (P - D)/P$, may be written $D = P(1 - S)$, and by comparing this equation with $y = mx$, it is clear that a regular D-scale on the

[1] Associate Professor of Mechanical Engineering, Sheffield Scientific School of Yale University. Mem. A.S M.E.

For presentation at the Annual Meeting, New York, November 30 to December 4, 1925, of THE AMERICAN SOCIETY OF MECHANICAL ENGINEERS. All papers are subject to revision. Figs. 4–6, 8–12, 21, 22, 25–28, 31–34, and 36–38 are reproduced on a greatly reduced scale from Design of Diagrams, by Hewes and Seward, copyright, 1923, McGraw-Hill Book Co., New York.

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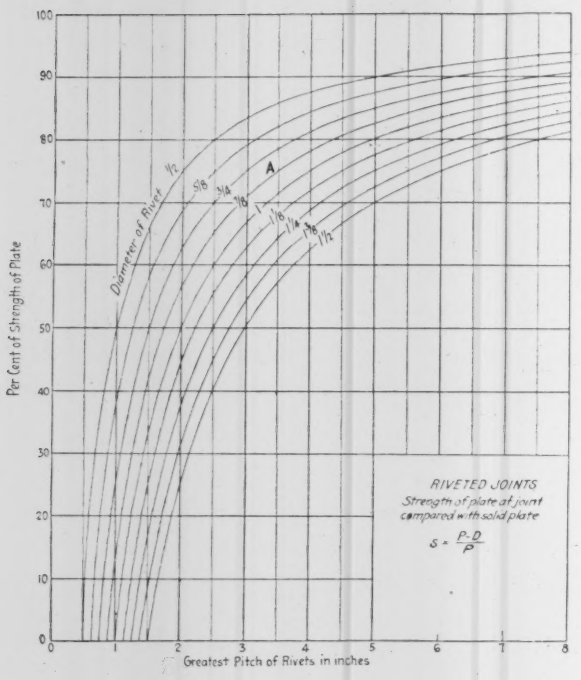


FIG. 4 PLOT OF FAMILY OF CURVES

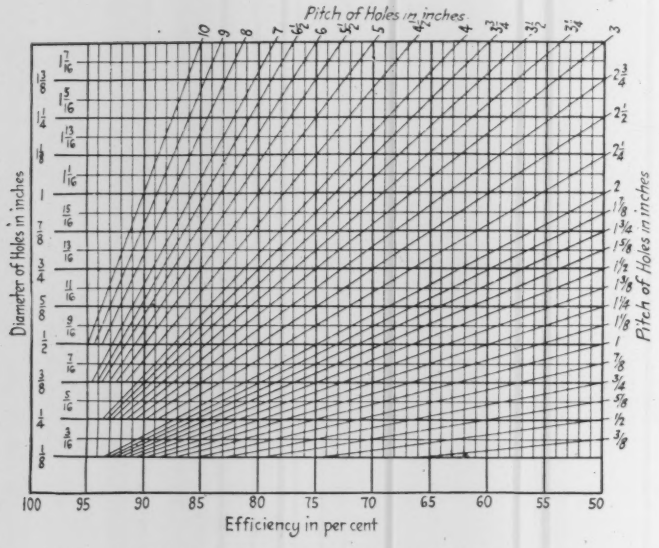


FIG. 5 RADIAL-TYPE STRAIGHT-LINE DIAGRAM

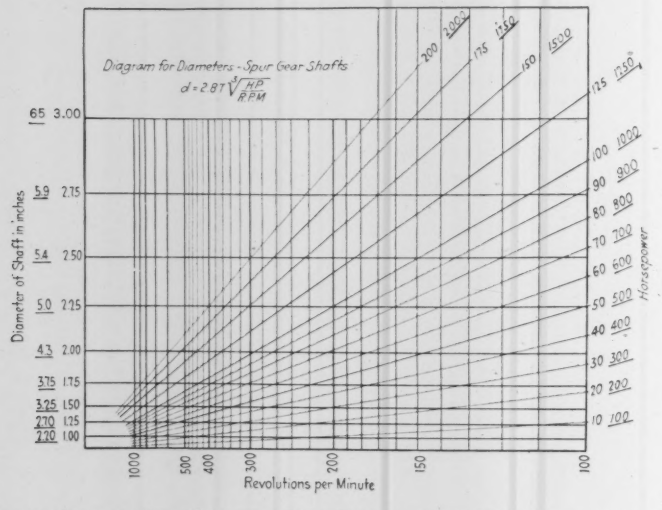


FIG. 6 RADIAL-TYPE STRAIGHT-LINE DIAGRAM WITH CUBE SCALE ON THE *Y*-AXIS AND RECIPROCAL SCALE ON THE *X*-AXIS

Y-axis, with a function scale $x = (1 - S)$ on the *X*-axis, will produce a family of radiating straight lines for P as in Fig. 5. Radial straight lines may best be plotted by connecting the origin to the points of the scale determined by them on some convenient line $X = K$.

The relation between diameter, horsepower, and r.p.m. of a steel shaft

$$d = 2.87\sqrt[3]{\frac{\text{hp.}}{\text{r.p.m.}}}$$

may be written

$$d^3 = (2.87)^3(\text{hp.})\,\frac{1}{\text{r.p.m.}}$$

or

$$y = m \cdot x$$

Thus with a cube scale on the *Y*-axis for d and a reciprocal scale on the *X*-axis for r.p.m., we have radial straight lines for hp. as in Fig. 6. Two corresponding sets of values are included for d and hp.

One of the simplest diagrams to illustrate the use of parallel

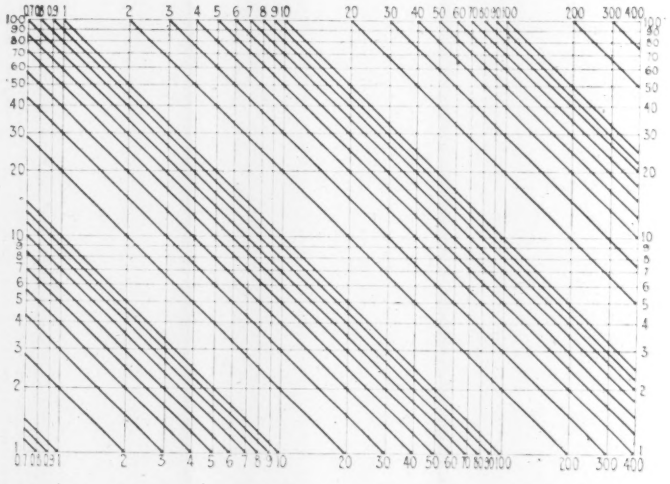


FIG. 7 DIAGRAM ILLUSTRATING USE OF PARALLEL STRAIGHT-LINE SYSTEMS

straight-line systems may be made on an ordinary piece of logarithmic cross-section paper. If the corresponding numbers on the axes are connected by 45-deg. lines (as in Fig. 7), the result is a diagram for multiplication and division. If

$$A \cdot B = C$$

we may write

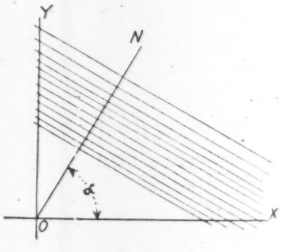


FIG. 8

$$\log A + \log B = \log C$$

The scales $x = \log A$ and $y = \log B$ are furnished on the axes, and

$$x + y = \log C$$

It will be evident that the system of 45-deg. lines determines a logarithmic scale on a line perpendicular to them (the normal), and that the scale factor of this scale is different from the scale factors used on the axes. The scale factor for the third set of parallel lines must always take the consequences of the choices made for the scale factors of the first two systems. In general, a set of parallel straight lines may be best drawn by first plotting the scale on the normal. In Fig. 8 any one of the parallel lines may be determined from the normal form of the general equation

$$x \cos \alpha + y \sin \alpha = p$$

where p is the distance from the origin to the given parallel line. In order to identify any equation $Ax + By + C = 0$ with the normal form it is necessary to divide through by $\pm\sqrt{A^2 + B^2}$,

with the sign of the radical opposite to the sign of C; or

$$\frac{A}{\pm\sqrt{A^2+B^2}}x + \frac{B}{\pm\sqrt{A^2+B^2}}y + \frac{C}{\pm\sqrt{A^2+B^2}} = 0$$

Then $$\cos\alpha = \frac{A}{\pm\sqrt{A^2+B^2}}$$

$$\sin\alpha = \frac{B}{\pm\sqrt{A^2+B^2}}$$

and the scale on the normal is

$$\frac{C}{\pm\sqrt{A^2+B^2}}$$

The formula

$$H = 0.38\frac{V^{1.86}}{D^{1.25}}$$

giving the friction head H in ft. per 1000 ft. of water flowing in a pipe D ft. in diameter with a velocity of V ft. per sec. may be written:

$$\log H + 1.25\log D = 1.86\log V + \log 0.38$$

If we plot $x = \log H$ and $y = 1.25\log D$, we have

$$x + y - (1.86\log V + \log 0.38) = 0$$

Dividing by $+\sqrt{2}$ gives

$$\frac{1}{\sqrt{2}}x + \frac{1}{\sqrt{2}}y - \frac{1.86\log V + \log 0.38}{\sqrt{2}} = 0$$

Comparison with $x\cos\alpha + y\sin\alpha - p = 0$, shows that the normal to the V-lines makes an angle of 45 deg. with the axes as in Fig. 9. Plotting the V-scale is best done by observing that the divisor $\sqrt{2}$ really transfers the points on the normal scale from the normal to the X-axis.

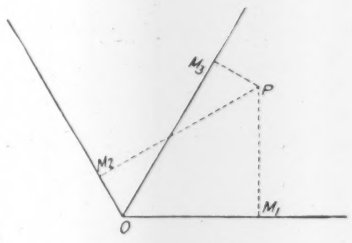


FIG. 10

The unit log scale, such as is used for the H-scale, is to be increased in length in the ratio 1.86:1.00 and set back a distance log 0.38. The origin is at the intersection of $D = 12$ in. and $H = 1.00$.

If the axes are taken at 120 deg. we have the "hexagonal diagram." In Fig. 10 it can be proved that $OM_3 = OM_1 + OM_2$, so that we have a geometrical adding machine. (OM_3 is the bisector.) If OM_1, OM_2, OM_3 are segments of function scales the diagram will solve equations in the sum form, but the scale factors on the three axes must be identical. Fig. 11 illustrates the use of this principle in connection with the last example. The equation is

$$(\log H - \log 0.38) + (1.25\log D) = (1.86\log V)$$

Diagrams for equations of four or more variables can often be made by separating the equation in such a way that two variables appear on each side of the equality sign. A "dummy" Δ may then be introduced and, in theory, two separate diagrams are constructed with Δ-scales identical. These diagrams may then be superimposed and a compact diagram results. For example, the equation of Fig. 12

$$d = \sqrt[3]{\frac{5.1\,F(L+\sqrt{L^2+R^2})}{f_s}}$$

may be written

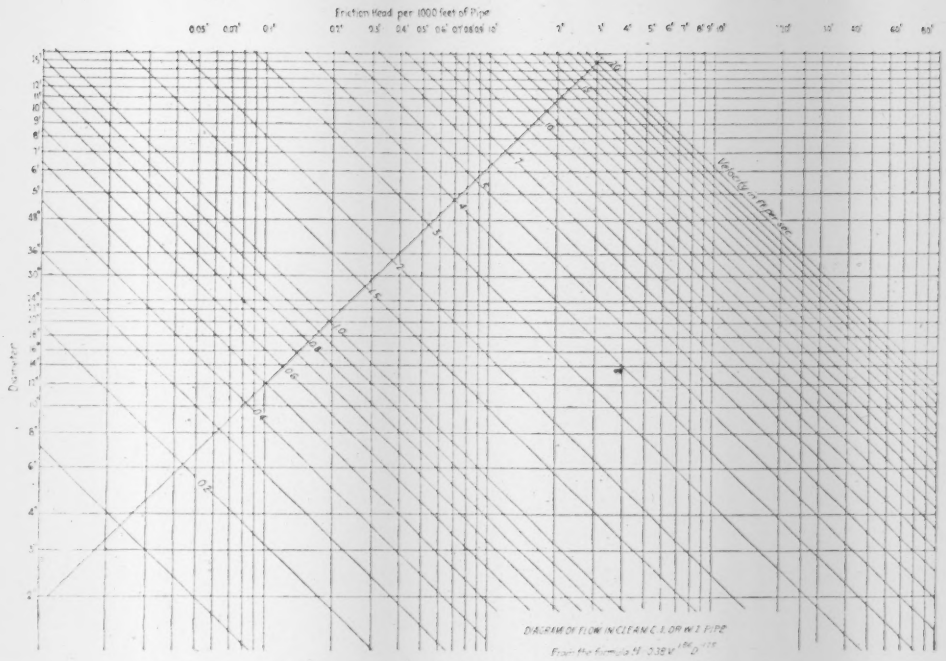


FIG. 9 PARALLEL STRAIGHT-LINE DIAGRAM FOR FLOW OF WATER IN PIPES

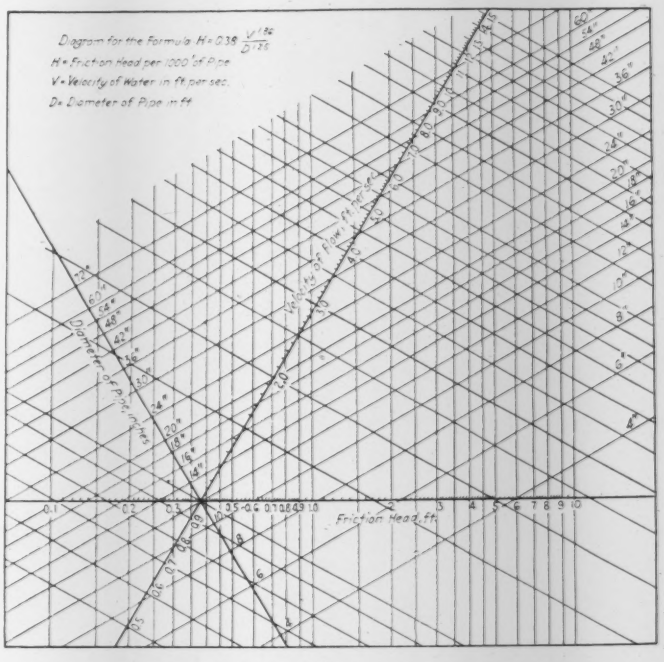


Diagram for the Formula H = 0.38 V^1.86 / D^1.25
H = Friction Head per 1000 ft of Pipe
V = Velocity of Water in ft per sec.
D = Diameter of Pipe in ft
Diameter of Pipe in Ins.
Friction Head, ft.

FIG. 11 HEXAGONAL DIAGRAM FOR FLOW OF WATER IN PIPES

$$d^3f_s = \Delta = 5.1\,F(L + \sqrt{L^2+R^2}).$$

If the parenthesis is called x, the radial lines for F in the figure are part of a separate diagram for

$$\Delta = 5.1Fx$$

with the dummy scale Δ on the Y-axis and determining the series of horizontal straight lines. The geometrical construction for obtaining the value of the parenthesis is explained on the diagram.

The lines for f_s at the left are really part of a separate diagram for $\Delta = d^3f_s$ or $y = xm$. The same dummy scale Δ together with a cube scale on the X-axis for d completes that part of the diagram. The two separate diagrams may be combined in any convenient way as long as the Δ-scales are identical and in horizontal alignment.

As an example of type 3 in the Cartesian diagrams, consider the cubic equation

$$X^3 + PX + Q = 0$$

given P and Q, required X. If $x = P$ and $y = Q$ are plotted on the OX and OY-axes, respectively, the equation becomes

$$Xx + y + X^3 = 0$$

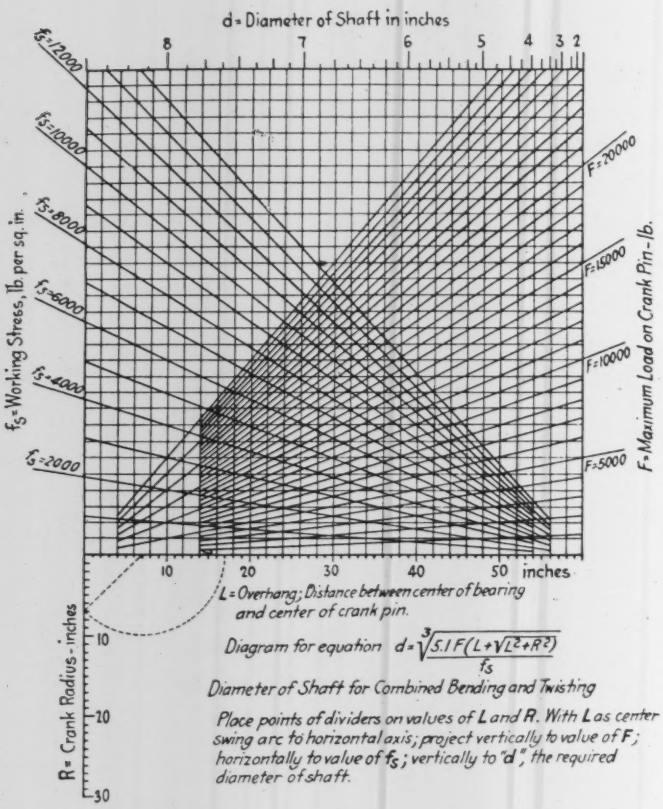

d = Diameter of Shaft in inches
8 7 6 5 4 3 2
fs=12000
fs=10000
fs=8000
fs=6000
fs=4000
fs=2000
fs = Working Stress, lb per sq. in.
F=20000
F=15000
F=10000
F=5000
F = Maximum Load on Crank Pin - lb.
10 20 30 40 50 inches
L = Overhang; Distance between center of bearing and center of crank pin.
R = Crank Radius - inches
10
20
30
Diagram for equation $d=\sqrt[3]{\frac{5.1F(L+\sqrt{L^2+R^2})}{f_s}}$
Diameter of Shaft for Combined Bending and Twisting
Place points of dividers on values of L and R. With L as center swing arc to horizontal axis; project vertically to value of F; horizontally to value of f_s; vertically to "d", the required diameter of shaft.

FIG. 12 TWO DIAGRAMS SUPERIMPOSED FOR EQUATIONS OF FOUR OR MORE VARIABLES

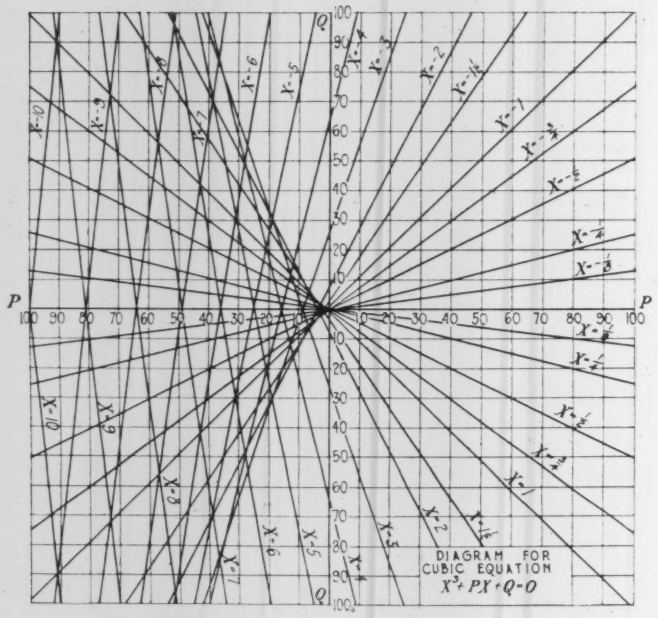


FIG. 13

which is similar to

$$Ax + By + C = 0$$

Each value of X substituted in this equation produces the equation of a different straight line, as in Fig. 13. The method would be no different if functions of P and Q happened to appear in the original equation. The scales for these functions would be plotted on the axes, instead of the regular scales. (See also Fig. 21.)

In considering Cartesian diagrams up to this point it will be noted that two of the three straight-line systems have been parallel systems, parallel to one of the coördinate axes, in each case. The most general conception of a Cartesian diagram would result in producing a diagram with no system parallel to a coördinate axis, as shown in Fig. 14. In general, when three straight lines L_1, L_2, L_3

$$\begin{aligned} (L_1) \quad & A_1x + B_1y + C_1 = 0 \\ (L_2) \quad & A_2x + B_2y + C_2 = 0 \qquad E_1 \\ (L_3) \quad & A_3x + B_3y + C_3 = 0 \end{aligned}$$

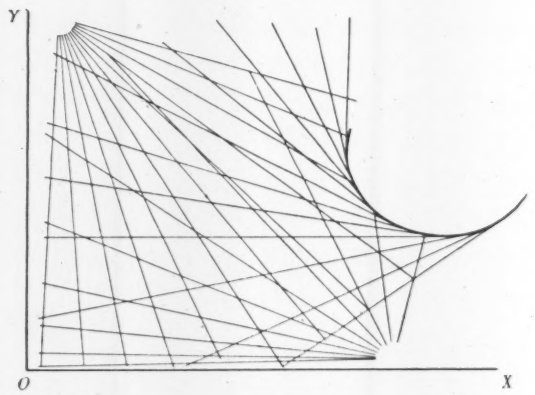


FIG. 14 DIAGRAM WITH NO SYSTEM PARALLEL TO A COÖRDINATE AXIS

meet at a point, as in Fig. 15, the coefficients from their equations satisfy the condition

$$\begin{vmatrix} A_1 & B_1 & C_1 \\ A_2 & B_2 & C_2 \\ A_3 & B_3 & C_3 \end{vmatrix} = 0 \qquad D_1$$

This determinant is merely a convenient method of writing

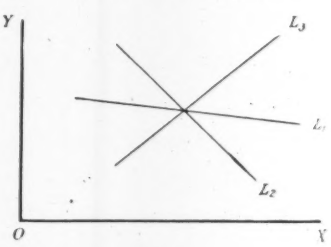


FIG. 15

$$A_1B_2C_3 + A_2B_3C_1 + A_3B_1C_2 - A_3B_2C_1 - A_2B_1C_3 - A_1B_3C_2 = 0$$

which can be thought of thus: Rewrite the first two columns to the right as in Fig. 15a, making the six combinations as shown

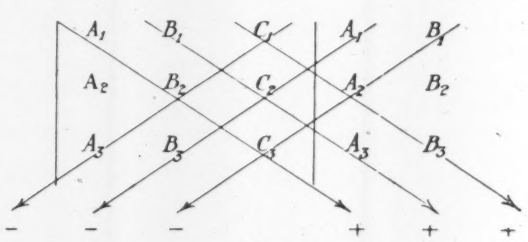


FIG. 15a

by the arrows, and apply the signs as indicated. (This method does not apply to a fourth-order determinant.)

If an engineering formula, a function of three variables, $F(z_1z_2z_3) = 0$ can be written in the determinant form,

$$\begin{vmatrix} f_1(z_1) & g_1(z_1) & h_1(z_1) \\ f_2(z_2) & g_2(z_2) & h_2(z_2) \\ f_3(z_3) & g_3(z_3) & h_3(z_3) \end{vmatrix} = 0$$

it is only necessary to consider these nine elements as the coefficients given in determinant D_1 above and write the three equations

$$\begin{aligned} f_1(z_1)x + g_1(z_1)y + h_1(z_1) &= 0 \\ f_2(z_2)x + g_2(z_2)y + h_2(z_2) &= 0 \\ f_3(z_3)x + g_3(z_3)y + h_3(z_3) &= 0 \end{aligned}$$

similar to equations E_1 and we have the equations of three straight-line systems which, when plotted, will constitute a diagram for the given formula. Note that only functions of a single variable

may be included in each line of the determinant, a limitation which will be later removed. As a simple illustration of this general theory we may check up on the method used in the last example, Fig. 13. Given $X^3 + PX + Q = 0$, which may be written:

$$\begin{vmatrix} 1 & 0 & -P \\ 0 & 1 & -Q \\ X & 1 & X^3 \end{vmatrix} = 0$$

resulting in the three straight-line systems:

$$\begin{aligned} (1)\quad & 1x + 0y - P = 0 \\ (2)\quad & 0x + 1y - Q = 0 \\ (3)\quad & Xx + 1y + X^3 = 0 \end{aligned}$$

or

$$\begin{aligned} (1)\quad & x = P \\ (2)\quad & y = Q \\ (3)\quad & Xx + y + X^3 = 0 \end{aligned}$$

which are the three straight-line systems used in Fig. 13.

It should be further noted that if a formula cannot be written as a determinant, no diagram can be made for it. It is difficult to formulate rules for writing formulas in the determinant form, and still more difficult to develop criteria to apply to formulas for the purpose of ascertaining whether they may or may not be written in the determinant form.

Sometimes an equation may appear in such a form that a judicious choice of the angle between the coördinate axes will simplify the plotting. Soreau shows some clever diagrams in which a family of curves become circles, using this method.[1]

Alignment Diagrams

It is established in analytical geometry that the area of a triangle $P_1P_2P_3$ on the YOX-plane is, in terms of the coördinates of P_1, P_2 and P_3:

$$\text{Area} = {}^1/_2(x_1y_2 + x_2y_3 + x_3y_1 - x_3y_2 - x_1y_3 - x_2y_1)$$

or

$$A = {}^1/_2\begin{vmatrix} x_1 & y_1 & 1 \\ x_2 & y_2 & 1 \\ x_3 & y_3 & 1 \end{vmatrix}$$

What would happen if the area of the triangle were reduced to zero? Either the three points $P_1P_2P_3$ would coincide or would become a straight line. For the purposes of this paper the last condition is extremely useful. If we set the above value of A equal to zero, we have a statement of a condition which places three points in alignment, as follows:

$$\begin{vmatrix} x_1 & y_1 & 1 \\ x_2 & y_2 & 1 \\ x_3 & y_3 & 1 \end{vmatrix} = 0 \qquad \text{[I]}$$

If a given engineering formula of three variables, $F(z_1z_2z_3) = 0$ could be arranged in a form similar to the above determinant, with functions of each of the three variables in their respective rows, a general type form would result as follows:

$$\begin{vmatrix} f_1 & g_1 & 1 \\ f_2 & g_2 & 1 \\ f_3 & g_3 & 1 \end{vmatrix} = 0 \qquad \text{[II]}$$

where f_1g_1 are any functions of z_1 (variable No. 1); f_2, g_2 are any functions of z_2 (variable No. 2); and f_3, g_3 any functions of z_3 (variable No. 3). This determinant [II] is so similar in form to [I] that we may write the comparison:

$$\left.\begin{aligned} x_1 &= f_1 & y_1 &= g_1 & &1. \\ x_2 &= f_2 & y_2 &= g_2 & &2. \\ x_3 &= f_3 & y_3 &= g_3 & &3. \end{aligned}\right\} \qquad \text{[III]}$$

[1] Mémoires de la Société des Ingénieurs Civils de France, Bulletin d'aout 1901, pp. 223–238.

Equations 1, 2 and 3 are the parametric equations of three curves. That is, if x_1 happened to be $r \sin \alpha$ and $y_1 = r \cos \alpha$; (r being a constant) the successive substitutions of values of α in the equations for x_1 and y_1 would, when plotted, give rise to a circle of radius r, center at the origin. The equation of this circle $x_1^2 + y_1^2 = r^2$ may be obtained from x_1 and y_1 in this case by squaring and adding.

But if any three pairs of values of the variables satisfy the given formula (Equation [II]), they must lie on a straight line across the three curves plotted from Equation [III], because of the similarity in form of the two determinants [I] and [II]. Thus we have a new type of diagram whose construction is based on the following simple rule of two steps:

1 Write the given formula as a determinant in form similar to [II].

2 Interpret the determinant and plot the resulting scales from the parametric Equations [III], choosing proper useful ranges for the variables.

A very simple alignment diagram for three variables may be constructed thus:

1 Copy the lower scales (C and D) of the slide rule on two parallel lines, any distance apart, opposite one another as in Fig. 16.

2 Add a copy of the upper scales (A and B) on a line halfway between these scales, Fig. 17.

This third scale has a factor just half as great as the other two. A thread or straight line anywhere across the picture gives the solution of $MN = P$. To prove this, rewrite the equation in the form

$$\log M + \log N = \log P$$

then

$$\begin{vmatrix} -1 & \log M & 1 \\ +1 & \log N & 1 \\ 0 & \frac{1}{2}\log P & 1 \end{vmatrix} = 0$$

which, expanded, is $-\log N + \frac{1}{2}\log P + \frac{1}{2}\log P - \log M = 0$, which checks. This is in the form [I] above, hence

$$\begin{aligned} x_1 &= -1 & y_1 &= \log M & &1. \\ x_2 &= +1 & y_2 &= \log N & &2. \\ x_3 &= 0 & y_3 &= \frac{1}{2}\log P & &3. \end{aligned}$$

which, plotted, becomes Fig. 17, with the P-scale on the Y-axis. The vertical unit of measurement is independent of the horizontal unit. The angle YOX may have any value.

If regular scales had been used in place of the logarithmic scales the result would be a graphical adding machine. It would be good exercise for the reader to arrange the equation $R + S = T$ as a determinant and justify the construction of such a diagram. Then try letting $x_1 = 1$, $x_2 = 2$, and find out what relations develop in the unit lengths of the three scales.

Alignment diagrams for equations of three variables may be classified as:

a Three parallel straight scales

b Two parallel scales and one oblique straight scale

c Three oblique straight scales

1 intersecting at a point

2 forming a triangle

d Two straight parallel scales and a curved scale

e Three curved scales

and other cases. To each type of diagram there corresponds a particular form of equation, and certain rules of procedure in plotting may be associated with each type. The author feels that the reader gains greater power in constructing alignment diagrams by manipulating and interpreting the determinant rather than in trying to algebraically persuade a given formula to assume one of the limited number of standard forms for which simple rules may be formulated. Cartesian diagrams are of necessity forced to fall into a few formal types. Alignment diagrams may occur in an endless number of combinations of straight and curved scales,

depending on the particular form of determinant written. Once having the formula written as a determinant, one familiar with the rules of determinants may so change its form as to rearrange or redesign the diagram and produce the most economical or accurate form.

The case of three parallel straight scales is so frequent and so generally applicable to many engineering formulas that it will be thoroughly analyzed. Similar methods of analysis may be applied

FIG. 16 CONSTRUCTION OF A SIMPLE ALIGNMENT DIAGRAM FOR THREE VARIABLES—FIRST STEP

FIG. 17 CONSTRUCTION OF A SIMPLE ALIGNMENT DIAGRAM FOR THREE VARIABLES—COMPLETION

to other types of diagrams. This form can only arise from an equation in the sum form $A + B = C$. A product may be written in logarithmic form to bring the equation to the sum form, thus often accommodating fractional powers which would otherwise be very awkward to handle.

If it is desired to have A vary from A_1 to A_2 and B vary from B_1 to B_2 while the length of these scales is limited to L inches, the equations are:

$$L_A = M_1(A_2 - A_1)$$
$$L_B = M_2(B_2 - B_1)$$

from which the scale factors M_1 and M_2 may be computed. If the limits for A and B are correctly chosen, the limits of C will be automatically taken care of. The choice of M_1 and M_2 has now determined the relative distances between the scales. From the sketch of conditions of alignment in Fig. 18 we may set up the determinant

$$\begin{vmatrix} -\delta_1 & M_1A & 1 \\ +\delta_2 & M_2B & 1 \\ 0 & M_3C & 1 \end{vmatrix} = 0$$

Expanding this and comparing with the original formula $A + B = C$, the following relations appear:

$$\frac{M_1}{M_2} = \frac{\delta_1}{\delta_2} \qquad M_3 = \frac{M_1M_2}{M_1 + M_2}$$

Hence the rule: To construct a diagram of three parallel straight scales for a formula $A + B = C$,

I Determine the scale factors M_1 and M_2 to give the desired range of variables A and B.

II Draw the two parallel scales for A and B at any convenient distance apart.

III Divide the distance between the A- and B-scales into $\delta_1 + \delta_2$ parts and place the C-scale parallel to the A- and B-scales so that

$$\frac{\delta_1}{\delta_2} = \frac{M_1}{M_2}$$

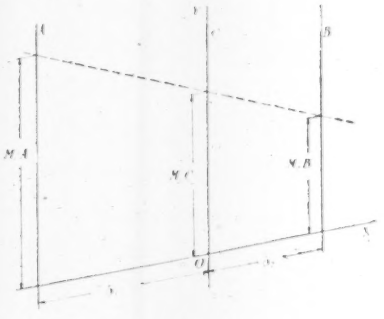


FIG. 18

IV Compute $M_3 = \dfrac{M_1M_2}{M_1 + M_2}$ and measure the scale for C with this factor, locating it by alignment of any known value of C, for given values of A and B.

(The formula for M_3 now explains the presence of the $^1/_2$ in the first alignment diagram.) For example, the formula

$$C = \frac{D^2}{256}\left[\frac{h^2}{20} + 2.6h - 4\right]$$

as derived by K. J. T. Eckblaw for the capacity C in tons of a silo of D ft. diameter and h ft. high, may be written

$$2 \log D + \log [f(h)] = \log C + \log 256$$

If D is to vary from 10 to 25 and its scale is to be 9.5 in. in length,

$$M_D = \frac{9.5}{2 \log 24 - 2 \log 10} = 11.6, \text{ say, } 12 \text{ in.}$$

If h is to vary from 20 to 120 and its scale is to be 9.5 in. in length,

$$M_h = \frac{9.5}{\log [\text{max.}] - \log [\text{min.}]} = 8$$

Then

$$\frac{\delta_1}{\delta_2} = \frac{M_D}{M_h} = \frac{12}{8} = \frac{3}{2}$$

and

$$M_3 = M_c = \frac{M_DM_h}{M_D + M_h} = \frac{3 \times 2}{3 + 2} = 1.2$$

The complete diagram is shown in Fig. 19.

It sometimes happens that two equations in four variables are given, when a diagram on which the alignment of four points is

possible. In three-stage air compression from pressure P_1 to pressure P_2 the ideal intercooler pressures are

$$P_a = \sqrt[3]{P_1^2 P_2}$$

$$P_b = \sqrt[3]{P_1 P_2^2}$$

The first equation may be written

$$\begin{vmatrix} -\delta_1 & 2M_1 \log P_1 & 1 \\ +\delta_2 & N_2 \log P_2 & 1 \\ 0 & \frac{3}{2} M_3 \log P_a & 1 \end{vmatrix} = 0$$

In order to have the P_1- and P_2-scales of the same length, let the scale factor $M_1 = \frac{1}{2}$ and $M_2 = 1$. Then the scale factor M_3 for the P_a-scale will be $\frac{1 \times 1/2}{1 + 1/2} = \frac{1}{3}$ and $\frac{\delta_1}{\delta_2} = \frac{1/2}{1} = \frac{1}{2}$, showing that all three scales are identical and the P_a scale is located at a point one-third of the distance from the P_1-scale to the P_2-scale. Similar quantities are obtained from the equation for P_b, with the result that the given equations may be solved by the curious arrangement of four equal logarithmic scales, equally spaced as in Fig. 20. A straight edge anywhere across the scales gives four values which satisfy the given pair of equations.

As an example leading to the use of curved scales, consider the cubic equation

$$z^3 + pz + q = 0$$

which may be written

$$\begin{vmatrix} \frac{z-1}{z+1} & \frac{-z^3}{z+1} & 1 \\ 1 & p & 1 \\ -1 & q & 1 \end{vmatrix} = 0$$

thus giving the three scales

1. $x = \frac{z-1}{z+1}$, $y = \frac{-z^3}{z+1}$
2. $x = 1$, $y = p$
3. $x = -1$, $y = q$

as drawn in Fig. 21. The origin is half-way between the zero of the p-scale and the zero of the q-scale. The curved scale for z is in two branches, the point —1 being at infinity. The existence of imaginary roots is clearly shown when the given values of p and q cause the straight edge to intersect the curved scale but once.

In the logarithmic mean-temperature-difference expression

$$d = \frac{T_1 - T_2}{\log_e \frac{T_1}{T_2}}$$

which is shown in Fig. 22 there is a curious symmetry which causes two curves to coincide. The determinant is

$$\begin{vmatrix} d & 0 & 1 \\ \frac{T_1}{\log_e T_1} & \frac{1}{\log_e T_1} & 1 \\ \frac{T_2}{\log_e T_2} & \frac{1}{\log_e T_2} & 1 \end{vmatrix} = 0$$

It is evident that the three curves are

$$x = d \qquad y = 0 \qquad 1.$$

$$x = \frac{T_1}{\log_e T_1} \qquad y = \frac{1}{\log_e T_1} \qquad 2.$$

$$x = \frac{T_2}{\log_e T_2} \qquad y = \frac{1}{\log_e T_2} \qquad 3.$$

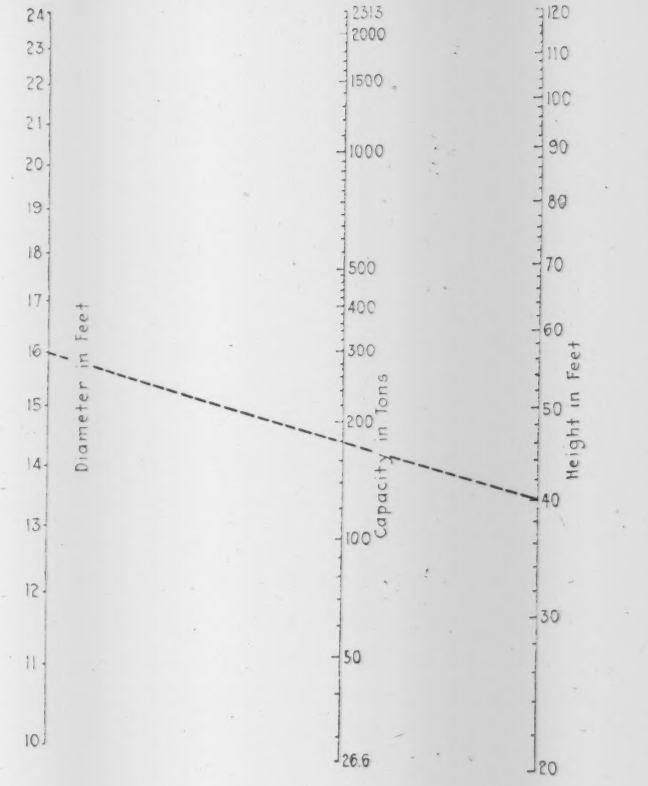


FIG. 19 ALIGNMENT DIAGRAM WITH THREE PARALLEL STRAIGHT SCALES FOR CAPACITY OF SILOS

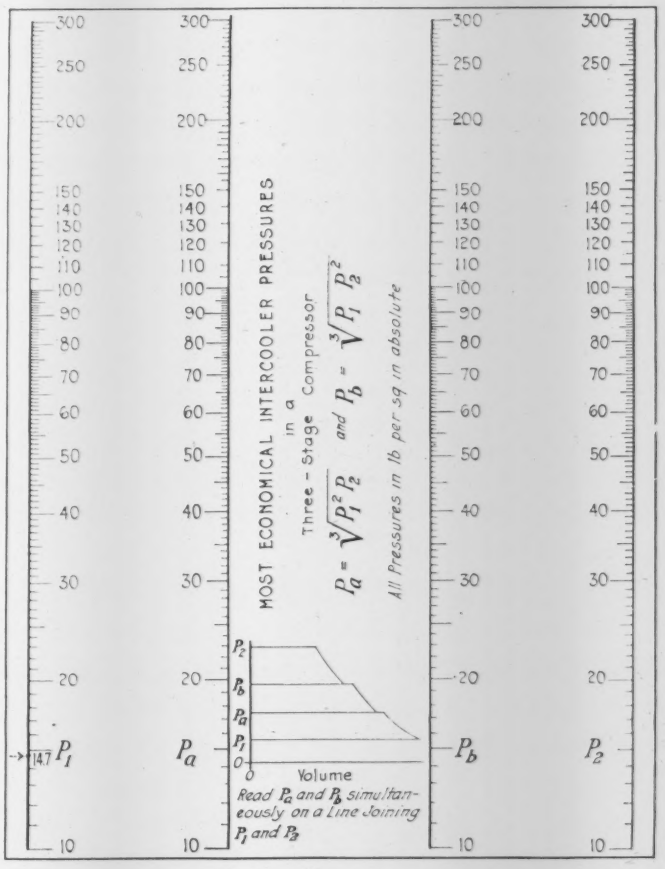

MOST ECONOMICAL INTERCOOLER PRESSURES
in a
Three - Stage Compressor
All Pressures in lb per sq in absolute
Volume
Read P_a and P_b simultaneously on a Line Joining P_1 and P_2
14.7

FIG. 20 ALIGNMENT DIAGRAM WITH FOUR EQUAL LOGARITHMIC SCALES

Be sure to bring this copy with you to the Annual Meeting

Eliminating T from either pair of equations results in finding $x = ye^{1/y}$ as the equation of the curved support for T_1 and T_2 referred to the d-scale as the Y-axis. Fig. 22 lacks some desirable features of accuracy when d is required, given T_1 and T_2, due to the cantilever action necessary in placing the straight edge.

Fig. 23 gives the solution for the expression

$$R = \frac{K}{2N}\left(1 + 2N - K^2\right)$$

and the meaning of the symbols is also indicated on the diagram. The determinant is

$$\begin{vmatrix} K^2 - 1 & \frac{1}{K} & 1 \\ 0 & \frac{1}{R} & 1 \\ 2N & 0 & 1 \end{vmatrix} = 0$$

It is hoped that when the reader looks at a determinant such as this, he not only sees the resemblance to

$$\begin{vmatrix} x_1 & y_1 & 1 \\ x_2 & y_2 & 1 \\ x_3 & y_3 & 1 \end{vmatrix} = 0$$

but is also able to develop a mental picture of the three scales and their characteristics. In this case x_1 and y_1 both are variable quantities, hence the K-scale is curved. It will be noted that the K- and R-scales have the same ordinates, which greatly facilitates the plotting. The presence of the two zeros in the determinant in opposite columns shows that the scales for R and N are straight and located on the OY- and OX-axes, respectively.

Lamé's formula for thick cylinders

$$m = \sqrt{\frac{R + P}{R - P}} - 1$$

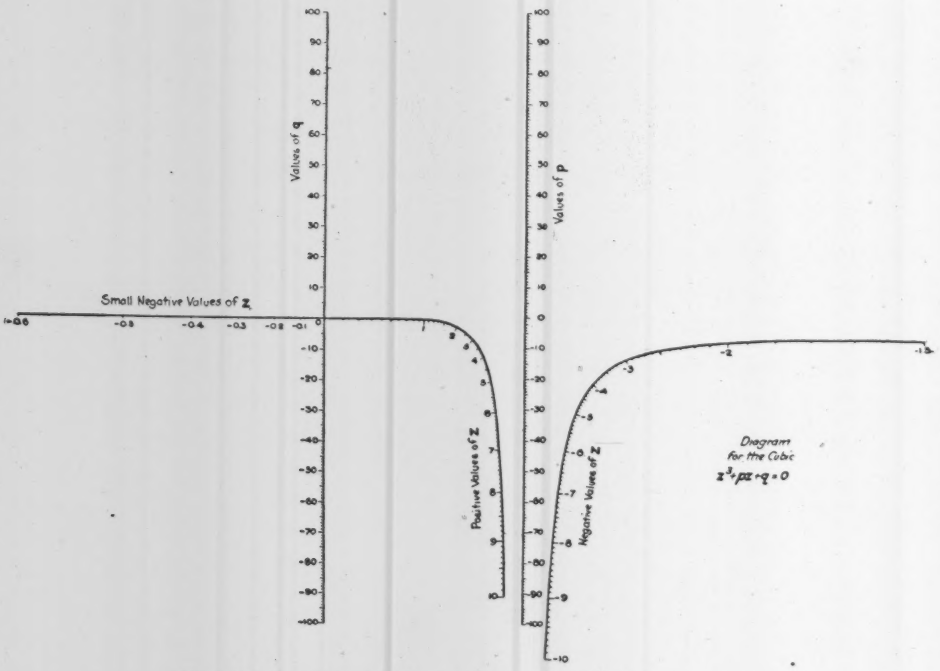


FIG. 21 DIAGRAM FOR CUBIC EQUATION WITH CURVED SCALES

as given in Fig. 24 may be written

$$-\frac{P}{R} + \frac{(1+m)^2 - 1}{(1+m)^2 + 1} = 0$$

or

$$\begin{vmatrix} 1 & \frac{P}{5} & \left(\frac{P}{5}\right)^2 \\ 1 & -\frac{10}{R} & \left(\frac{10}{R}\right)^2 \\ 1 & 0 & 2\frac{(1+m)^2 - 1}{(1+m)^2 - 1} \end{vmatrix} = 0$$

This is in the form to produce two conic scales and one straight-line scale, but the determinant has been so manipulated by Soreau

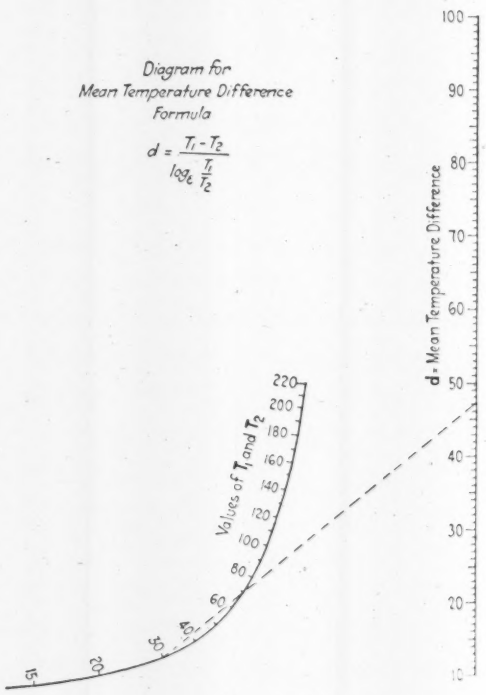


FIG. 22

as to change the two conic supports to the same circle, making the use of the laws of determinants. Rewriting, adding columns 1 and 3,

$$\begin{vmatrix} 1 & \frac{P}{5} & 1 + \left(\frac{P}{5}\right)^2 \\ 1 & -\frac{10}{R} & 1 + \left(\frac{10}{R}\right)^2 \\ 1 & 0 & 1 + 2\frac{(1+m)^2 - 1}{(1+m)^2 + 1} \end{vmatrix} = 0$$

and dividing by the last column,

$$\begin{vmatrix} \frac{1}{1 + \frac{P^2}{25}} & \frac{\frac{P}{5}}{1 + \frac{P^2}{25}} & 1 \\ \frac{1}{1 + \frac{100}{R^2}} & \frac{-\frac{10}{R}}{1 + \frac{100}{R^2}} & 1 \\ \frac{1}{1 + 2\frac{(1+m)^2 - 1}{(1+m)^2 + 1}} & 0 & 1 \end{vmatrix} = 0$$

from which the diagram is plotted. The origin is at the zero of the P-scale and the m-scale is on the X-axis.

The determinants so far considered have been similar to the general type

$$\begin{vmatrix} f_1 & g_1 & 1 \\ f_2 & g_2 & 1 \\ f_3 & g_3 & 1 \end{vmatrix} = 0$$

The Francis weir formula, $q = 3.33BH^{3/2}$

where q = discharge of water over weir, cu. ft. per sec.
B = breadth of weir, ft.
H = head on weir, ft. (sometimes corrected)

may be written

$$\begin{vmatrix} & B & 1 \\ 3.33H^{3/2} & 1 & 1 \\ 0 & 0 & 1 \end{vmatrix} = 0$$

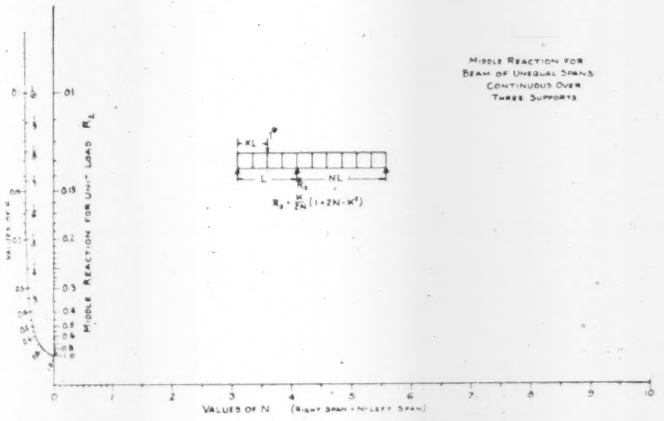


FIG. 23

Comparing this with the general alignment determinant

$$\begin{vmatrix} x_1 & y_1 & 1 \\ x_2 & y_2 & 1 \\ x_3 & y_3 & 1 \end{vmatrix} = 0$$

we observe that it expresses the alignment of the following three points:

1 Any point in the plane, determined by $x = q, y = B$
2 A point on a straight scale $x = 3.33H^{3/2}$ located on $y = 1$
3 The origin (0,0)

as given in Fig. 25.

Thus we have a diagram with one fixed point, through which a straight edge might be permanently attached and used to align any point on the H-scale with the desired point in the network. As this straight edge swings about the origin it is really generating a series of radial lines similar to a Cartesian diagram. This equation is but one illustration of a type of diagram with one fixed point. The importance of the last few examples to the reader is in accepting the interpretation of alignment as given by the determinant.

The use of a dummy scale in order to handle equations of more than three variables as explained in the discussion of Cartesian diagrams (Fig. 12) is also available with alignment diagrams. The same methods apply. The formula

$$T = 0.2618\,\frac{LFD}{S}$$

may be used in Fig. 26 as a simple illustration of these methods. Writing it

$$\log T + \log S = \log \Delta = \log F + \log L + \log D + \log 0.2618$$

makes it possible to construct at once a diagram for

$$\log T + \log S = \log \Delta$$

A second dummy is necessary in order to reduce the number of variables in

$$\log \Delta = \log F + \log L + \log D + \log 0.2618$$

from four to three. This may conveniently be taken as P, the product of L and D, so that

$$\log \Delta = \log F + \log P + \log 0.2618$$

which makes it possible to add the P- and F-scales to the diagram. Scale P need not necessarily be graduated if scales for D and L are added to the diagram from the relation $\log P = \log D + \log L$.

If this formula had appeared with exponents giving powers or

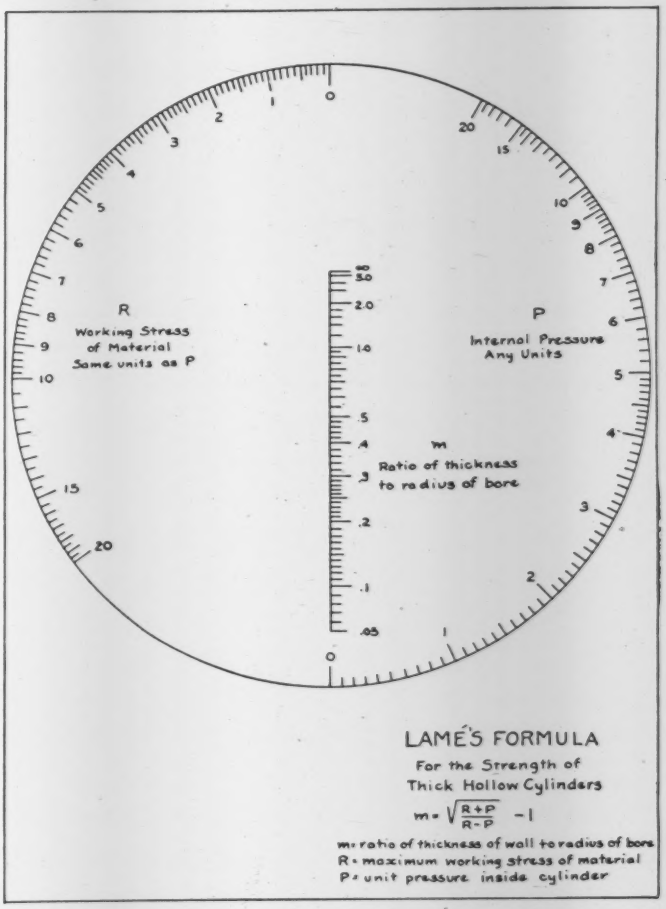

R
Working Stress of Material Same units as P
P
Internal Pressure Any Units
m
Ratio of thickness to radius of bore
LAMÉ'S FORMULA
For the Strength of Thick Hollow Cylinders
$m = \sqrt{\frac{R+P}{R-P}} - 1$
m = ratio of thickness of wall to radius of bore
R = maximum working stress of material
P = unit pressure inside cylinder

FIG. 24 DIAGRAM FOR LAMÉ'S THICK-CYLINDER FORMULA

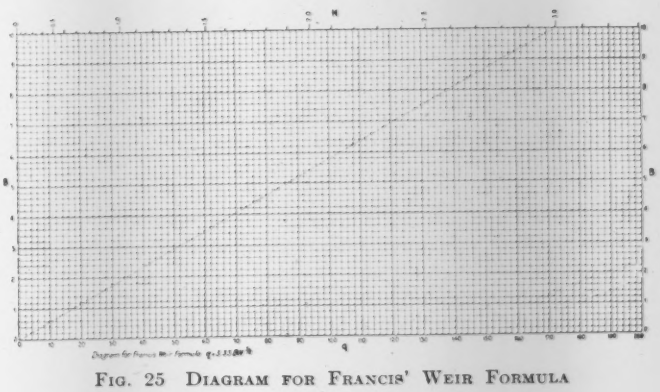


FIG. 25 DIAGRAM FOR FRANCIS' WEIR FORMULA

roots for every variable, the procedure would have been the same, with the exception that the unit lengths of the logarithmic scales would have varied, under the control of the exponents.

Certain types of equations in four variables may be handled by the use of parallel or perpendicular alignment. A piece of celluloid or tracing paper having parallel or perpendicular lines drawn on it becomes a necessary adjunct to the diagram. In analytical geometry when two lines P_1P_2 and P_3P_4 are parallel, the coördinates of P_1, P_2, P_3, and P_4 satisfy the equation

$$\frac{y_2 - y_1}{x_2 - x_1} = \frac{y_4 - y_3}{x_4 - x_3} = \frac{b}{a}$$

This equation may be considered as resulting from the elimination of a and b from the two equations:

$$\begin{vmatrix} a & b & 0 \\ x_1 & y_1 & 1 \\ x_2 & y_2 & 1 \end{vmatrix} = 0 \text{ and } \begin{vmatrix} a & b & 0 \\ x_3 & y_3 & 1 \\ x_4 & y_4 & 1 \end{vmatrix} = 0$$

If an equation of four variables may be put in the form

$$\frac{g_2 - g_1}{f_2 - f_1} = \frac{g_4 - g_3}{f_4 - f_3}$$

where g_2 is any function of variable No. 2, etc., the equation may be considered as resulting from the elimination of Δ from the two equations

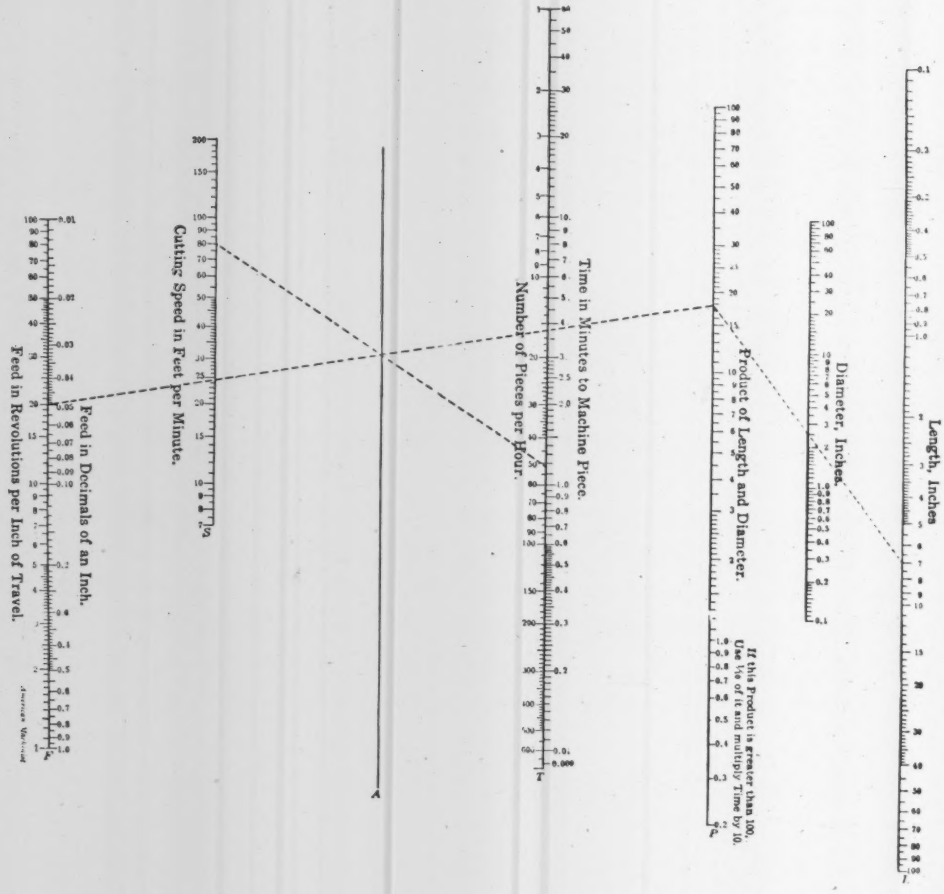

Feed in Decimals of an Inch.
Feed in Revolutions per Inch of Travel.
Cutting Speed in Feet per Minute.
Number of Pieces per Hour.
Time in Minutes to Machine Piece.
Product of Length and Diameter.
If this Product is greater than 100, Use ¹⁄₁₀ of it and multiply Time by 10.
Diameter, Inches.
Length, Inches

Fig. 26 Alignment Diagram for Employing Dummy Scales

$$\begin{vmatrix} 1 & \Delta & 0 \\ f_1 & g_1 & 1 \\ f_2 & g_2 & 1 \end{vmatrix} = 0 \text{ and } \begin{vmatrix} 1 & \Delta & 0 \\ f_3 & g_3 & 1 \\ f_4 & g_4 & 1 \end{vmatrix} = 0$$

and if diagrams are constructed for each of the determinants, the straight edge will have the same slope in both diagrams when solving the original equation of four variables. Resuming Lamé's formula and writing it

$$\frac{S + P}{S - P} = \frac{D^2 - 0}{0 + d^2}$$

or

$$\begin{vmatrix} 1 & \Delta & 0 \\ P & -P & 1 \\ S & S & 1 \end{vmatrix} = 0 \quad \begin{vmatrix} 1 & \Delta & 0 \\ -d^2 & 0 & 1 \\ 0 & D^2 & 1 \end{vmatrix} = 0$$

defines four scales:

1 $x = P$, $y = -P$, a regular scale on a straight 45-deg. line
2 $x = S$, $y = S$, a regular scale on a straight 45-deg. line
3 $x = -d^2$, $y = O_2$, a squared scale on $-OX$
4 $x = 0$, $y = D^2$, a squared scale on $+OY$

as in Fig. 27. Given three quantities, the fourth is obtained by a parallel ruler or tracing cloth.

In order to have two lines perpendicular, the slope of one must be the negative reciprocal of the other. To produce a diagram of perpendicular alignment we may rewrite the determinants

$$\begin{vmatrix} 1 & \Delta & 0 \\ f_1 & g_1 & 1 \\ f_2 & g_2 & 1 \end{vmatrix} = 0 \text{ and } \begin{vmatrix} 1 & -\frac{1}{\Delta} & 0 \\ g_3 & -f_3 & 1 \\ g_4 & -f_4 & 1 \end{vmatrix} = 0$$

or

$$\frac{g_2 - g_1}{f_2 - f_1} = -\frac{f_4 - f_3}{g_4 - g_3}$$

The last example may be reinterpreted and produce the diagram of Fig. 28 from these equations:

1 $x = P$ $y = -P$
2 $x = S$ $y = S$
3 $x = D^2$ $y = 0$
4 $x = 0$ $y = d^2$

in which the alignment requires two perpendicular lines scratched on a piece of celluloid or its equivalent.

Combinations of alignment diagrams involving perpendicular alignment with ordinary straight-line alignment are often useful to solve equations of four or five variables. The equation $L = d^2n^2K/257.4b$ of Fig. 29 is written $257.4\ L/n^2 = \Delta = d^2K/b$. A diagram for L, n, and Δ is readily constructed from

$$\begin{vmatrix} \log 257.4 - 2\log n & 0 & 1 \\ -\log \Delta & 1 & 1 \\ -\frac{1}{2}\log L & \frac{1}{2} & 1 \end{vmatrix} = 0$$

and the remaining part of the formula is written

$$\begin{vmatrix} 1 & \Delta & 0 \\ 1 & 2\log d & 1 \\ 0 & -\log K & 1 \end{vmatrix} = 0 \quad \begin{vmatrix} 1 & -\frac{1}{\Delta} & 0 \\ -\log & 1 & 1 \\ \log b & 0 & 1 \end{vmatrix} = 0$$

which produces the perpendicular alignment between K and d to the straight edge from b to Δ. Values of K are given in the supplementary diagram.

Fig. 30 shows a diagram for which the determinants are:

$$\begin{vmatrix} \frac{\Delta}{43} & 0 & 1 \\ \frac{R - 6.8}{43} & 1 & 1 \\ \frac{52.2d^{0.25}}{43(d^{1.25} - 1)} & \frac{d^{1.25}}{(d^{1.25} - 1)} & 1 \end{vmatrix} = \begin{vmatrix} 1 & \Delta_1 & 0 \\ 43 & 0 & 1 \\ 0 & -4.59\text{hp.} & 1 \end{vmatrix} =$$

$$\begin{vmatrix} 1 & -\frac{1}{\Delta_1} & 1 \\ 0 & \frac{1}{3f^{0.74}} & 1 \\ r + 2.1 & 0 & 1 \end{vmatrix} = 0$$

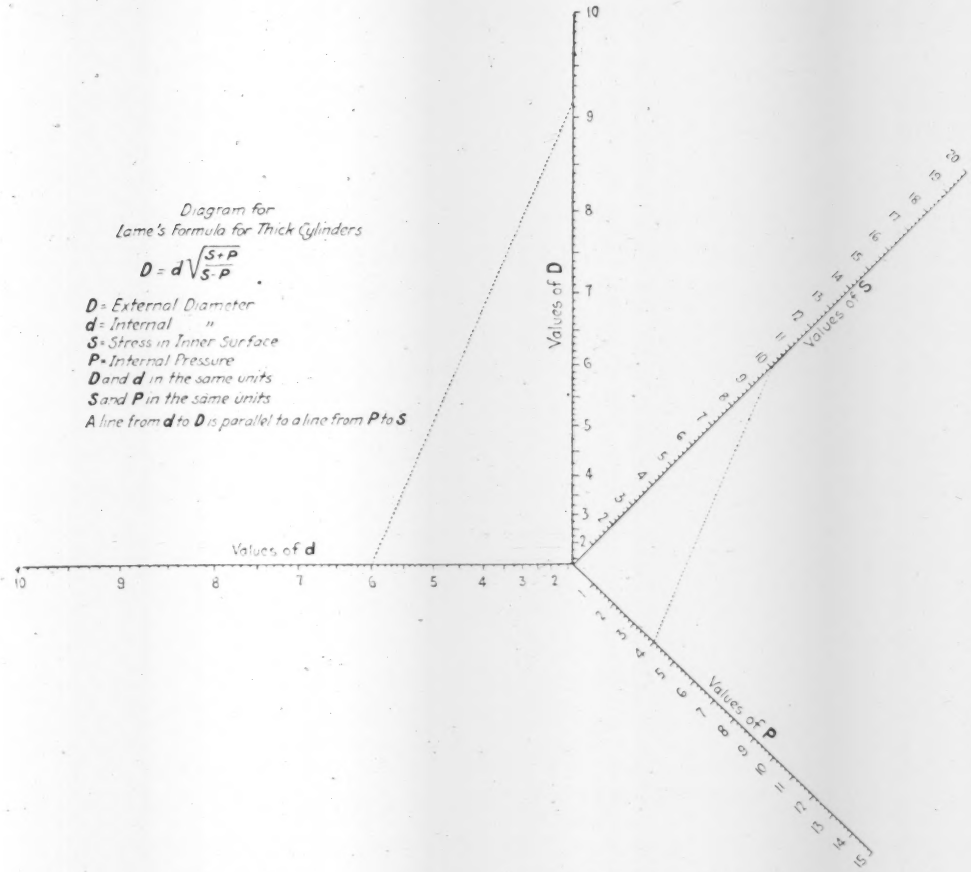


Diagram for
Lame's Formula for Thick Cylinders
$D = d\sqrt{\frac{S+P}{S-P}}$
D = External Diameter
d = Internal "
S = Stress in Inner Surface
P = Internal Pressure
D and d in the same units
S and P in the same units
A line from d to D is parallel to a line from P to S
Values of d
Values of D
Values of S
Values of P

FIG. 27 DIAGRAM FOR EQUATION WITH FOUR VARIABLES EMPLOYING PARALLEL ALIGNMENT

The d-scale is really a very flat curve with its point one at infinity.

It may often be possible to write an equation of four or more variables in determinant form, but functions of two variables will appear in the same line if the determinant is of the third order. The interpretation of such a determinant will lead to another type of diagram. If four variables are considered, and if $f_1 g_2$, etc., are any functions of the variables, the equation might be in the form

$$\begin{vmatrix} f_1 & g_1 & 1 \\ f_2 & g_2 & 1 \\ f_3 f_4 & g_3 g_4 & 1 \end{vmatrix} = 0$$

The first line defines a curved scale for variable No. 1

$$x = f_1$$
$$y = g_1$$

and the second line likewise, for variable No. 2

$$x = f_2$$
$$y = g_2$$

but the third line defines a set of points which can only be designated by the intersection points of two systems of curves:

$$x = f_3 f_4$$
$$y = g_3 g_4$$

as shown schematically in Fig. 31. There is still the alignment of three points, P_1, P_2 and P; the latter being a point in the plane at the intersection of a curve of the z_3-system with a curve of the z_4-system. A logical extension of this idea to an equation of six variables

$$\begin{vmatrix} f_1 f_2 & g_1 g_2 & 1 \\ f_3 f_4 & g_3 g_4 & 1 \\ f_5 f_6 & g_5 g_6 & 1 \end{vmatrix} = 0$$

would result in a scheme of alignment indicated in Fig. 32. Engineering formulas do not often appear in this symmetrical form. If a function is absent or a constant the diagram is immediately simplified.

The complete cubic equation

$$z^3 + a_1 z^2 + a_2 z + a_3 = 0$$

in which the three coefficients a_1, a_2, and a_3 are given, required the real roots z, may be rewritten as

$$\begin{vmatrix} 1 & a_2 & 1 \\ 0 & a_1 & 1 \\ \dfrac{z}{z^2 + z} & \dfrac{-(z^3 + a_3)}{z_2 + z} & 1 \end{vmatrix} = 0$$

The first two lines of the determinant define two ordinary straight scales at the edges of the diagram, Fig. 33,

$$x = 1 \qquad y = a_2$$
$$x = 0 \qquad y = a_1$$

The third line of the determinant involves functions of the two variables z and a_3. Since the value of x is free from a_3, containing only z,

$$x = \frac{z}{z^2 + z}$$

the curves for z are really a series of vertical straight lines spaced in accordance with values of x. Since a_3 appears as an increment in the numerator of y,

$$y = \frac{-(z^3 + a_3)}{z^2 + z}$$

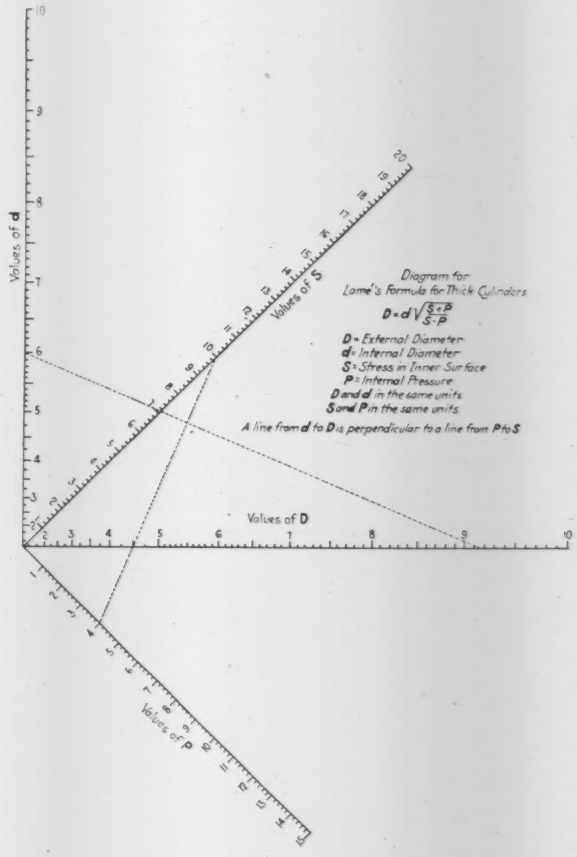

Values of d
Values of S
Values of D
Values of P
Diagram for
Lame's Formula for Thick Cylinders
$D = d\sqrt{\frac{S+P}{S-P}}$
D = External Diameter
d = Internal Diameter
S = Stress in Inner Surface
P = Internal Pressure
D and d in the same units
S and P in the same units
A line from d to D is perpendicular to a line from P to S

FIG. 28 DIAGRAM FOR EQUATION WITH FOUR VARIABLES EMPLOYING PERPENDICULAR ALIGNMENT

the successive a_3 curves will divide each vertical line into equal parts. It is only necessary to plot the curves for $a_3 = 10, 0,$ and -10 and divide each z-line into equal parts in order to plot the curves. The dotted line in the figure is set to solve

$$z^3 + 4z^2 - 4z + 0.5 = 0$$

and show that $z = 0.15$ and 0.69 because the curve $a_3 = 0.5$ cuts the vertical z-lines in these values.

Whenever x or y in the preceding type of determinant reduces to a constant or zero, a new type of scale results. It is called a

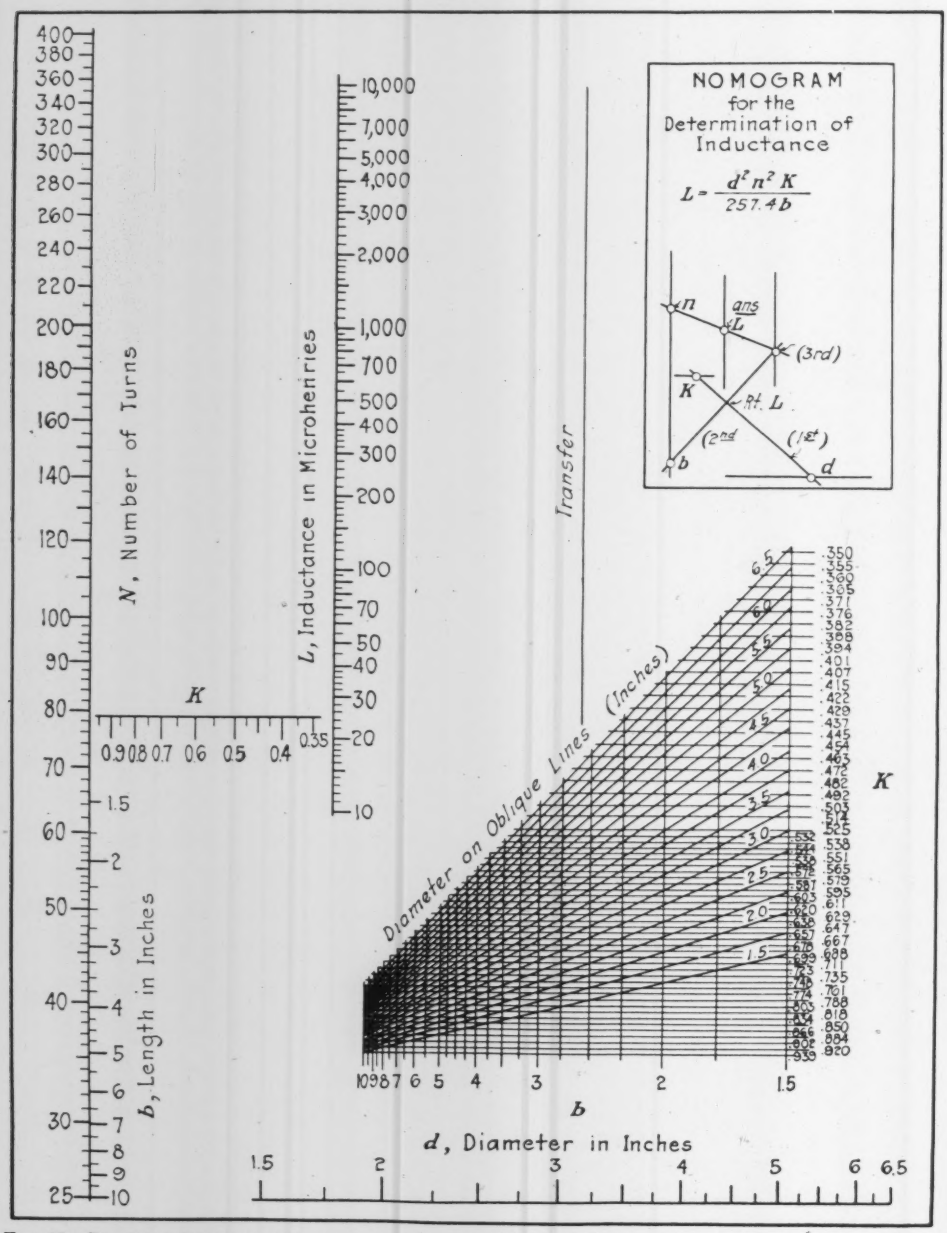


FIG. 29 DIAGRAM COMBINING PERPENDICULAR WITH ORDINARY STRAIGHT-LINE ALIGNMENT FOR SOLVING AN EQUATION WITH FOUR OR FIVE VARIABLES

"binary scale." In Fig. 34, consider OM. Each of the two systems of curves represents a range of values for the variables z_1 and z_2. A vertical line, as MP, traverses many intersection points of the two systems of curves and may be considered as bringing these intersection points down to point M. The length OM is then a function of the two variables z_1 and z_2, thus constituting a segment of a function scale for the two variables. OM is essentially not different from the dummy scale used in preceding forms of diagrams. As an example, consider

$$T = \frac{d^2 S \times 0.85P}{W}$$

as defined in Fig. 35. The determinant form is

$$\begin{vmatrix} \frac{T}{P} & 0 & 1 \\ 0.85d^2 & 1 & 1 \\ 0 & -\frac{S}{W} & 1 \\ & 1 - \frac{S}{W} & \end{vmatrix} = 0$$

The ratio T/P may be simplified by taking $P = 100$ and calling the result tractive power per 100 lb. boiler pressure (the resultant T to be multiplied by the actual boiler pressure). There are then two simple horizontal scales, one for T and one for d. The scale for S and W is a binary scale

$$x = 0$$

$$y = \frac{-\frac{S}{W}}{1 - \frac{S}{W}} = \frac{S}{W - S}$$

The alignment principle requires that the scale be located on $x = 0$ but other geometrical means are necessary to segregate W and S. We may choose a system of lines to represent one variable, say, W, as $x = W$, a series of vertical straight lines. Substituting x for W in y gives

$$y = \frac{s}{x - s}$$

defining a series of curves for S. These curves, together with the lines for W and the horizontal transfer lines constitute a binary scale on the OY-axis. Its purpose is to bring an intersection point to a line on the diagram which is part of the alignment scheme. The dotted line shows the procedure, given W, S and d, required T.

Practically every type of third-order determinant can be handled by methods already given, except the type where a variable appears more than once and in more than one row of the determinant. Consider a determinant

$$\begin{vmatrix} f_1 f_2 & g_1 g_2 & 1 \\ f_2 f_3 & g_2 g_3 & 1 \\ f_3 f_1 & g_3 g_1 & 1 \end{vmatrix} = 0$$

There are functions of three variables present, but they are not segregated into rows, as heretofore. Applying the alignment theory we may produce schematically the conditions shown in Fig. 36 from the equations:

1 $x = f_1 f_2$ $y = g_1 g_2$

2 $x = f_2 f_3$ $y = g_2 g_3$

3 $x = f_3 f_1$ $y = g_3 g_1$

Each of the three variables is represented by two different sets of curves. It will be possible only by some adjustment of the straight edge to make it line up with corresponding values of the variables. Such a diagram is called by Dr. L. I. Hewes an "alignment diagram with adjustment." It may appear unnecessarily complicated at first sight, but inasmuch as it removes the limitation imposed on the earlier types of determinants, it admits a whole

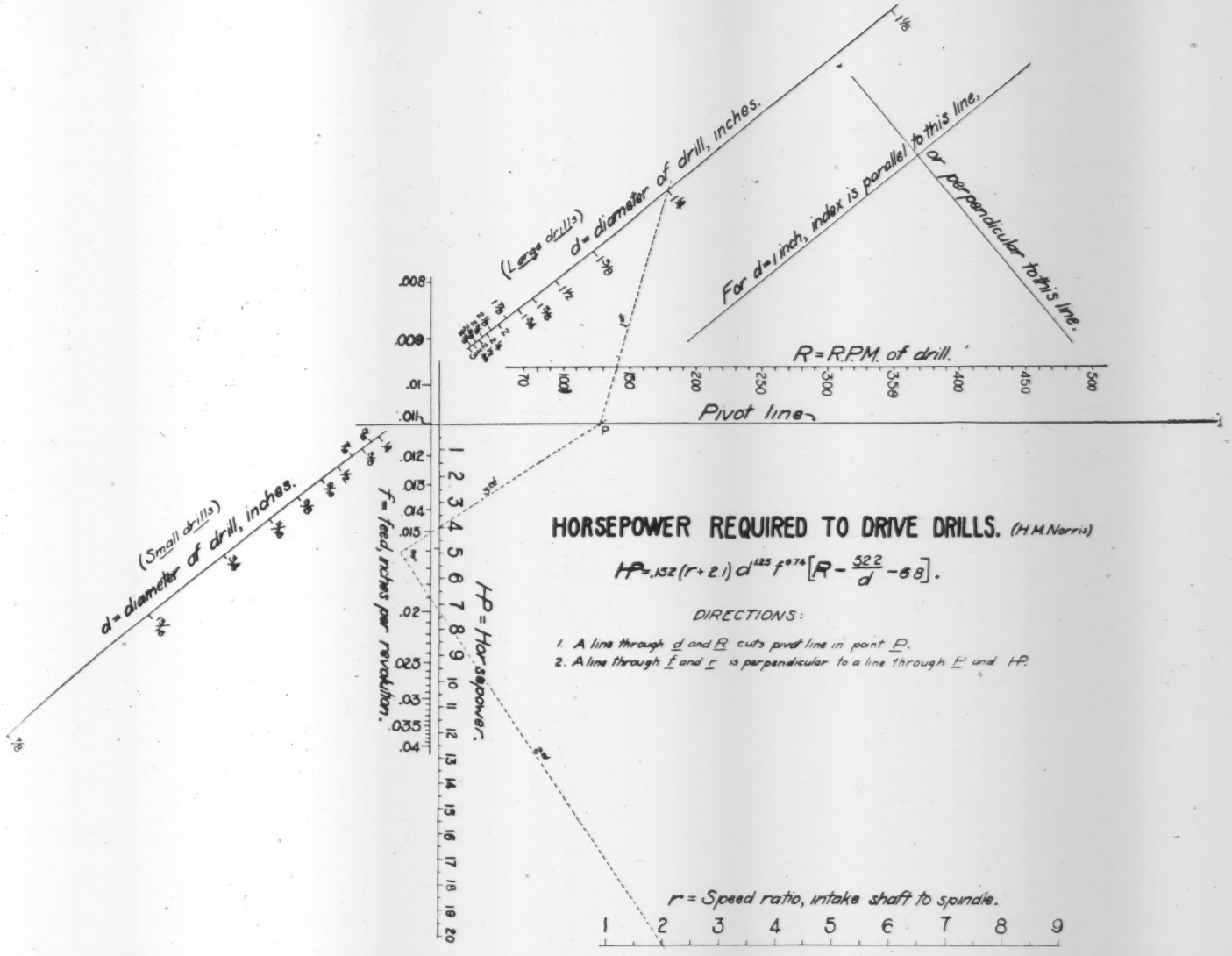

d = diameter of drill, inches.
(Large drills)
For d = 1 inch, index is parallel to this line,
or perpendicular to this line.
R = R.P.M. of drill.
Pivot line
(Small drills)
d = diameter of drill, inches.
f = feed, inches per revolution.
HP = Horsepower.
HORSEPOWER REQUIRED TO DRIVE DRILLS. (H.M.Norris)
$HP = .132(r + 2.1)\,d^{1.23} f^{0.74}\left[R - \frac{322}{d} - 68\right]$.
DIRECTIONS:
1. A line through d and R cuts pivot line in point P.
2. A line through f and r is perpendicular to a line through P and HP.
r = Speed ratio, intake shaft to spindle.

FIG. 30 DIAGRAM COMBINING PERPENDICULAR WITH ORDINARY STRAIGHT-LINE ALIGNMENT FOR SOLVING AN EQUATION WITH FOUR OR FIVE VARIABLES

great class of equations into the field of graphical representation which have hitherto been impossible of solution. Sometimes the adjustment reduces to zero because of the absence of some variable.

The open-belt problem has been solved by many approximate methods, but in Fig. 37 is given a solution which is accurate in principle and inaccurate only in so far as the limitations of scale and drafting may limit it. The equation is

$$L = R(\pi + 2\theta) + r(\pi - 2\theta) + 2C\cos\theta$$

It will be necessay to take C as 100 and express R, r and L in proportion to C as 100.

Rewriting

$$\begin{vmatrix} 1 & R & 1 \\ -1 & r & 1 \\ \frac{2}{\pi}\theta & \frac{L}{2} - 100\cos\theta & 1 \end{vmatrix} = 0$$

$$\sin\theta = \frac{R - r}{C}$$

Using a horizontal unit ten times the vertical unit, we may plot the three systems:

1 $x = 10$, $y = R$, a regular scale at the right

2 $x = -10$, $y = r$, a regular scale at the left

and 3 $x = \frac{20}{\pi}\theta$, $y = \frac{L}{2} - 100\cos\theta$

Eliminating θ from Equation 3 gives the equation of the L-curves as

$$y = \frac{L}{2} - 100\cos\frac{\pi}{20}x$$

As x is a function of θ only, it is also a function of $(R-r)$ only, so that a series of vertical lines for $(R-r)$ is next drawn in. The curves for L are symmetrical to the Y-axis and are translated curves. When L is required, there is no adjustment of the straight edge necessary, but the real open-belt problem is to find a new pair of radii to use the same L. This may be accomplished by trial until R, r, line up with the intersection of $R-r$ and the required L, all in terms of $C = 100$.

Dean Arthur M. Greene, Jr., gives[1] a formula for mean temperature difference

$$T = \left[\frac{n(T_1 - T_2)}{T_1^n - T_2^n}\right]^{\frac{1}{1-n}}$$

which may be written

$$\begin{vmatrix} 0 & \frac{T^{(1-n)}}{n} & 1 \\ 1 & T_1 & T_1^n \\ 1 & T_2 & T_2^n \end{vmatrix} = 0$$

One column of this determinant must be made to read unity in all three positions in order to apply the methods of the alignment dia-

[1] In Heat Engineering, McGraw-Hill Book Co.

Be sure to bring this copy with you to the Annual Meeting

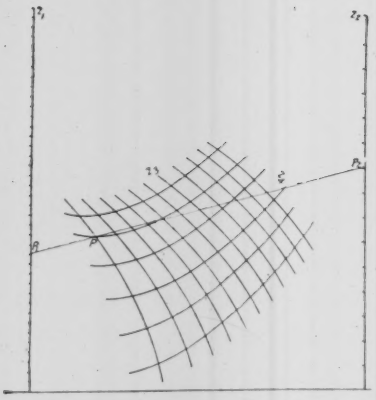


FIG. 31

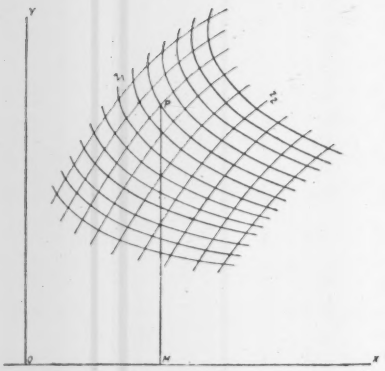


FIG. 34

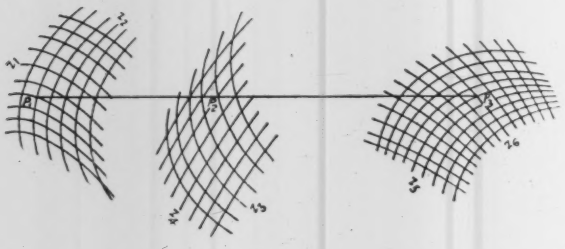


FIG. 32

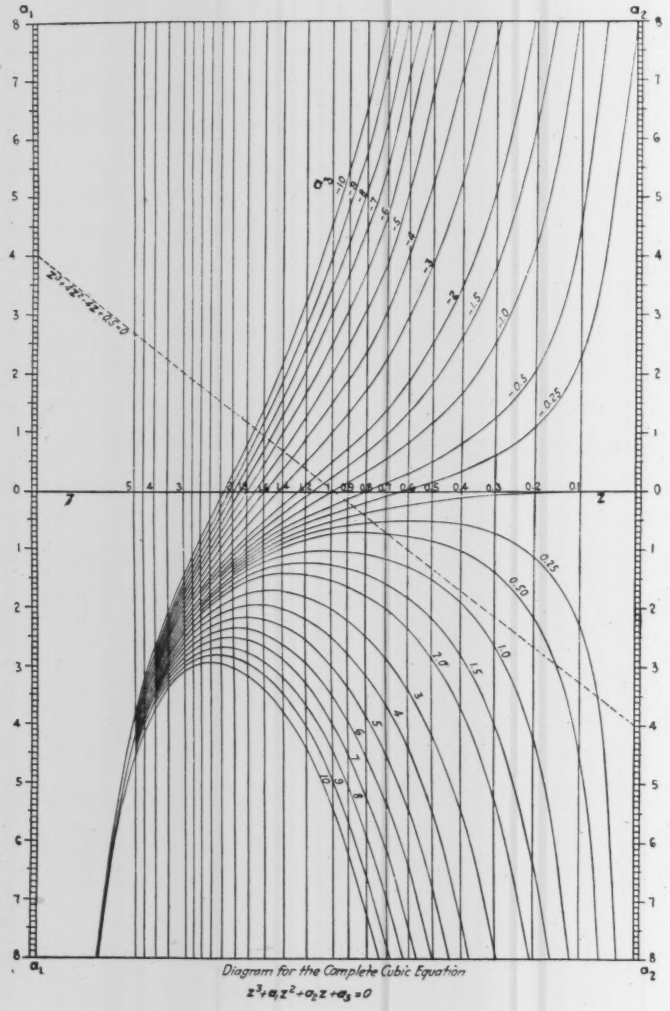


FIG. 33

gram. If the determinant is divided by the last column, there will be some reciprocals of powers of T_1 and T_2 which will be difficult to compute and plot. If the first column is multiplied by some number, say, 20, and added to the second column to produce a new third column we may divide by this new third column and have

$$\begin{vmatrix} 1 & \dfrac{n}{T^{(1-n)}} & 1 \\ \dfrac{T_1}{T_1+20} & \dfrac{T_1{}^n}{T_1+20} & 1 \\ \dfrac{T_2}{T_2+20} & \dfrac{T_2{}^n}{T_2+20} & 1 \end{vmatrix} = 0$$

The exponent n enters every line, but as T is usually the unknown, there will be no adjustment of the straight edge necessary. The intersection of n and T_1 will be

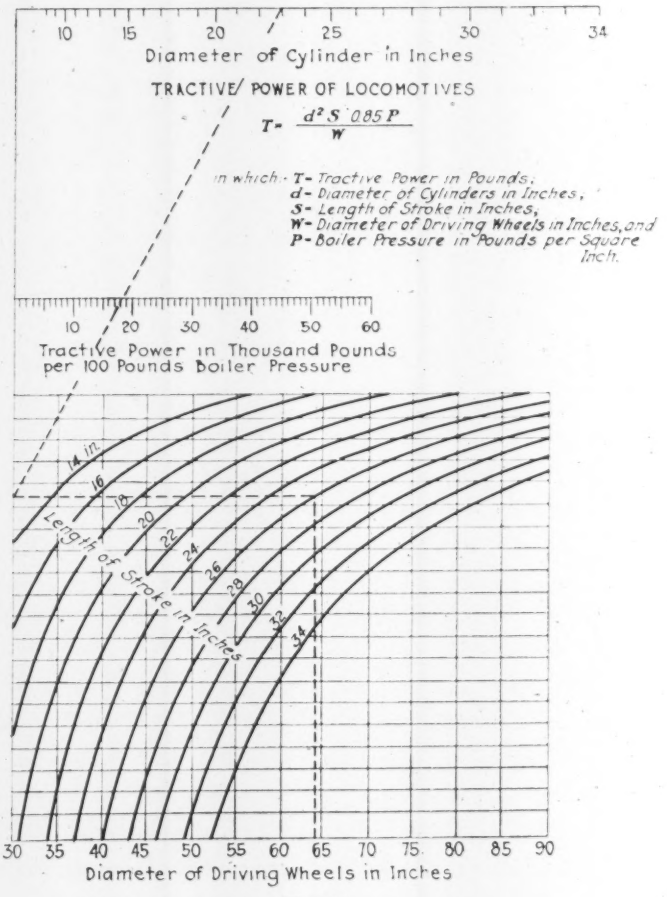

Diameter of Cylinder in Inches
TRACTIVE POWER OF LOCOMOTIVES
$T=\frac{d^2S\ 0.85P}{W}$
in which T- Tractive Power in Pounds,
d- Diameter of Cylinders in Inches,
S- Length of Stroke in Inches,
W- Diameter of Driving Wheels in Inches, and
P- Boiler Pressure in Pounds per Square Inch.
Tractive Power in Thousand Pounds per 100 Pounds Boiler Pressure
Length of Stroke in Inches
Diameter of Driving Wheels in Inches

FIG. 35

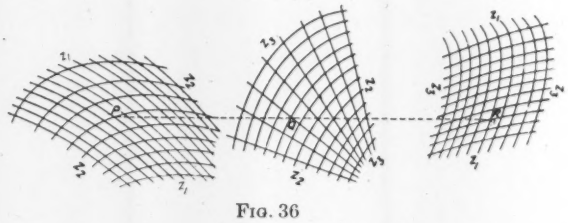


FIG. 36

lined up with the intersection of n and T_2, then T will be read at the intersection with the same value of n previously used at T_1 and T_2.

The equations

$$1 \quad x = 1 \qquad y = \frac{n}{T^{(1-n)}}$$

$$2 \quad x = \frac{T_1}{T_1+20} \qquad y = \frac{T_1{}^n}{T_1+20}$$

$$3 \quad x = \frac{T_2}{T_2+20} \qquad y = \frac{T_2{}^n}{T_2+20}$$

show that scale No. 1 is a binary scale on $x = 1$. In devising a plan to segregate n and T of this binary scale, note that

$$x = \frac{T_1}{T_1 + 20}$$

$$x = \frac{T_2}{T_2 + 20}$$

are sure to be the scales used for T_1 and T_2, and the equations define a system of parallel vertical lines. It would be economical to adopt these same lines as part of the binary scale for n and T, with the result that

$$y = \frac{n(1 - x)^{(1-n)}}{(20x)^{(1-n)}}$$

are the curves for n in the binary scale. Fig. 38 shows the diagram. The dotted lines show the solution of $n = 0.44$, $T_1 = 70$, $T_2 = 15$, from which $T = 37$. Find the intersections of the n curve for 0.44 (of the T_1 and T_2 group) with $T_1 = 70$ and with $T_2 = 15$. Prolong the alignment of these intersections to the right of the numbered scales (5.7 in this case). The numbers on this scale and on the left-hand scale are merely to carry a horizontal alignment from any point such as 5.7 in this case. Horizontally opposite 5.7 at the intersection with the other n-curve for 0.44 will be found the value of T desired, namely, 37.

ACKNOWLEDGMENT

The author has not found it possible to give individual credit in the body of this paper to those who have helped in the development of the various examples used. He is especially grateful to his associate, Dr. L. I. Hewes; to many former students whose solutions of problems have been used in this paper, including Prof. P. G. Laurson, Mr. L. F. Southwick and others. Their intelligent and sympathetic interest in and out of class has been a real inspiration.

In his office files the author has a collection of diagrams probably unequaled for variety and interest, produced by graduate students in this work since 1908. In writing this paper the problem has been to select only the most general and economical to reproduce. The question of accuracy cannot be discussed in the present brief treatment. To conserve space some of the illustrations have been photographically reduced from originals twenty inches square, but the reader must not infer that such cramped scales or congested areas are necessary or desirable on a working diagram. If any member of the Society is interested in further study along these lines, the author will cheerfully aid him.

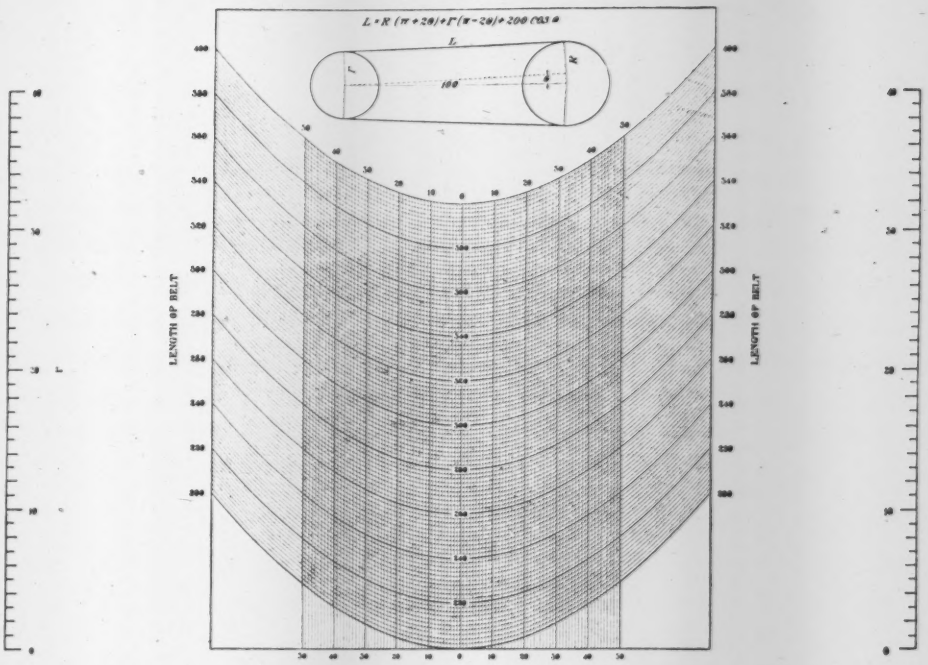


FIG. 37 DIAGRAM SOLVING THE OPEN-BELT PROBLEM

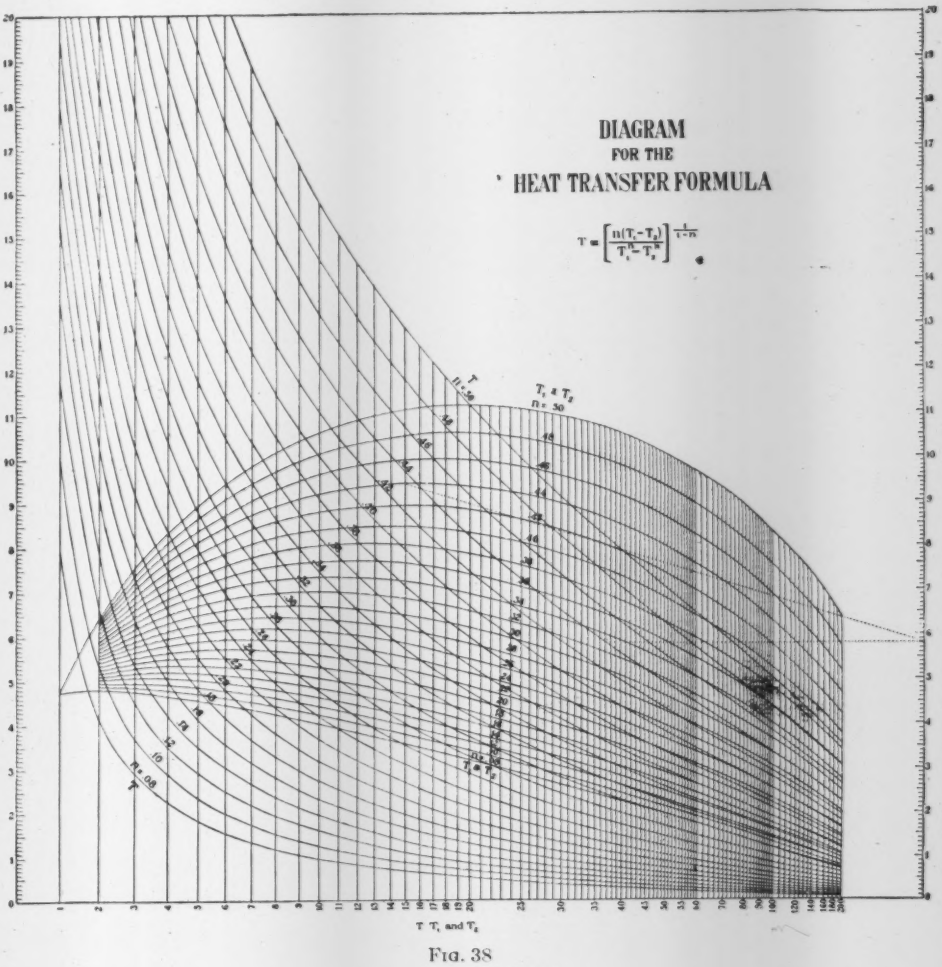


FIG. 38

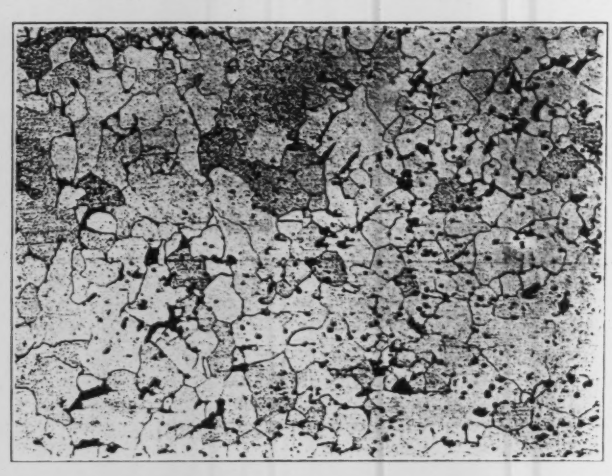
Bolts for Use in Power-Plant Construction

Steel vs. Wrought Iron and Screw Stock—Best Carbon Content for Bolts—Tentative Specifications for High-Temperature-Alloy Bolting Material

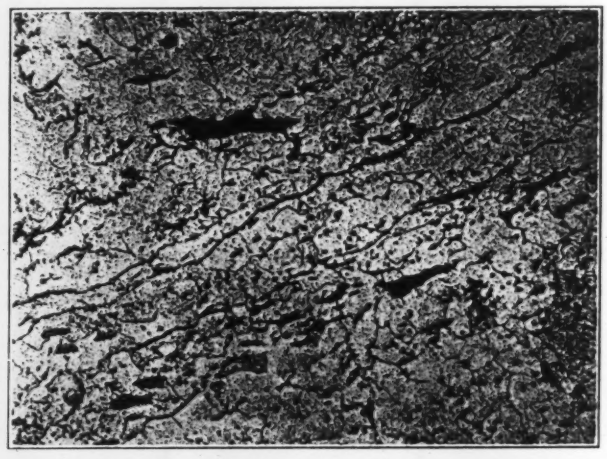
By WILLIAM P. WOOD,[1] ANN ARBOR, MICH.

THE stock for bolts and nuts is just now beginning to receive careful consideration from designing engineers. Hitherto bolts and nuts have been looked upon as one of the connecting links in construction work, but a link of no importance from a physical-property standpoint. The only requirement was that they should possess proper conformance to certain dimensional features.

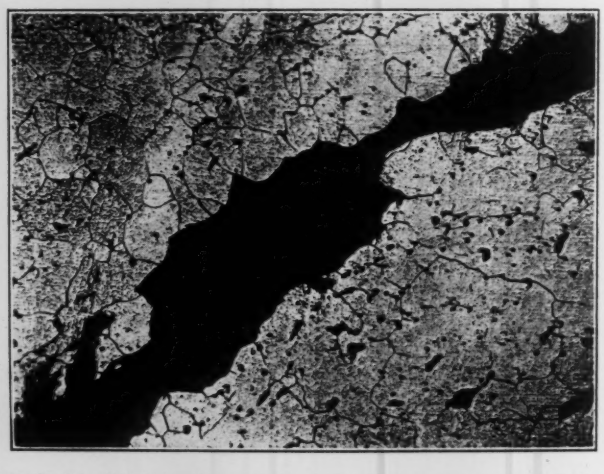
Such a viewpoint is in process of a revolutionary change. Engineers, especially designing engineers for large power companies, are today giving careful scrutiny to the character of the bolts and nuts used in their plants. This paper covers the findings from one such study.



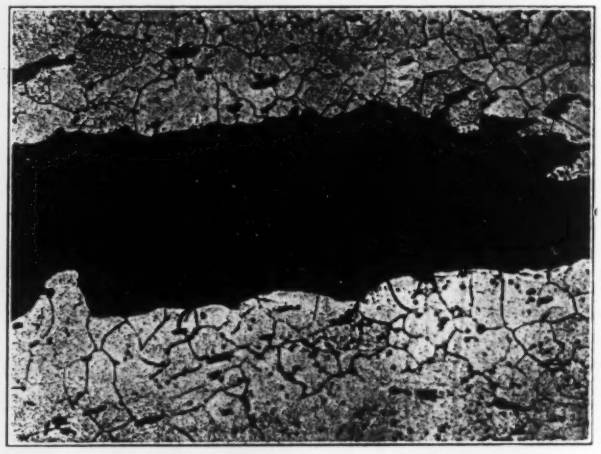
No. 1 Cross-section of bolt No. 1. Left-hand portion fairly free from slag; right-hand portion carries much slag.



No. 3 Longitudinal section of bolt No. 3. Note the streaked and non-uniform quality of the metal.



No. 2 Longitudinal section of bolt No. 2. Note the large slag inclusion.



No. 4 Longitudinal section of bolt No. 4. Note the large slag inclusion.

FIG. 1 PHOTOMICROGRAPHS OF SECTIONS OF FOUR WROUGHT-IRON BOLTS
(Etch, 4 per cent picric-nitric acid mixture in alcohol; magnification, × 100.)

STEEL VS. WROUGHT IRON

One of the conditions which became apparent after a little investigation, was the indiscriminate use of steel and wrought iron as bolting material. It is unquestioned that there are many instances where this may be done without serious consequences, but is the construction of high-pressure steam lines one of these instances? Granted that the use of wrought iron may be feasible in some instances, it should at least be stipulated that the wrought iron be of as good quality as possible. That this is not always the case may be shown by citing a few examples. The photomicrographs shown in Fig. 1 represent sections from two wrought-iron steam-line bolts. Two things stand out in these four photomicrographs: first, the presence of some enormous slag inclusions (the black portions), and second, the extreme non-homogeneity of the crystalline portions of the bolts. A good piece of wrought iron should have a structure similar to the left-hand portion of photomicrograph No. 1. Fig. 2 is a slightly enlarged view of the macroscopically etched longitudinal sections of these two bolts. Note the extreme fiber distortion, which has gone so far in some cases as to actually rupture the surface of the bolt. These two sections were taken immediately under the heads of the bolts. The structure shown here almost suggests that these bolts were "cold headed."

The physical properties of these two wrought-iron bolts are given in Table 1. While the ductility of these two bolts is acceptable, the elastic limit and tensile strength should be at least 50 per cent higher than they are.

[1] Assistant Professor of Metallurgical Engineering, University of Michigan.

Contributed by the Power Division for presentation at the Annual Meeting, New York, November 30 to December 4, 1925, of THE AMERICAN SOCIETY OF MECHANICAL ENGINEERS. All papers are subject to revision.

Be sure to bring this copy with you to the Annual Meeting

TABLE 1 PHYSICAL PROPERTIES OF TWO WROUGHT-IRON BOLTS

(Diameter, 0.505 in.; gage length, 2 in.)

Sample	Elastic limit, lb. per sq. in.	Maximum strength, lb. per sq. in.	Elongation, per cent	Reduction of area, per cent
Bolt No. 1.....	35,000	52,500	26.5	46.0
Bolt No. 2.....	41,500	58,500	18.0	44.5

Many instances were found where shanks of bolts were made from steel and nuts from wrought iron. One lot of bolts which were investigated were particularly interesting from this standpoint. The bodies of the bolts were made from a hardened 0.4 per cent carbon steel and possessed unusually high tensile properties coupled with good ductility. The nuts, on the other hand, were made from a poor quality of "bundled scrap." Photomicrograph No. 1 of Fig. 3 represents the structure of the shank of steel bolt No. 3. The structure is sorbitic, which means that it is fairly strong and tough. Photomicrographs Nos. 2, 3, and 4 of the same figure represent the structures of wrought-iron nuts Nos. 1, 2, and 3. As may be easily seen, the metal is made up of alternate layers of steel and wrought iron, the mottled-appearing portion being the steely part and the other the wrought iron. Such a condition is very undesirable on account of non-uniformity, and satisfactory physical properties could hardly be expected. Material of this nature is made by bundling wrought iron and steel scrap together, heating the bundles up to a welding temperature, and rolling the plastic mass out into merchant bars. Such stock is inferior to a good grade of wrought iron. It is quite customary to use a material in the nuts which is somewhat inferior to that used in the bodies of the bolts, but the use of such poor stock as in this case is certainly to be condemned.

TABLE 2 PHYSICAL TESTS OF 71 STEAM-LINE BOLTS

Specimen	Diameter, in.	Elastic limit, lb. per sq. in.	Maximum strength, lb. per sq. in.	Breaking point, lb. per sq. in.	Gage length, in.	Elongation, per cent, basis, 2 in.	Reduction of area, per cent	Material
1-H	0.501	31,400	51,700	44,000	2	31.0	46.4	W. I.
1-I	0.501	33,000	51,700	47,500	2	17.5	35.0	W. I.
1-O	0.502	34,800	56,000	50,000	2	16.5	35.0	W. I.
2-A	0.505	33,500	52,500	44,000	2	22.0	48.5	W. I.
2-C	0.501	35,500	53,300	43,600	2	32.5	51.0	W. I.
3-C	0.501	69,500	100,000	62,000	2	26.0	67.0	Steel
4-E	0.502	40,000	56,000	50,000	2	16.5	35.0	W. I.
4-K	0.502	35,000	53,000	47,000	2	24.0	47.0	W. I.
4-N	0.502	38,400	54,500	48,500	2	19.5	37.3	W. I.
5-A	0.376	32,450	52,200	45,400	1.5	23.75	45.0	W. I.
6-A	0.378	43,500	62,500	39,000	1.5	40.0	79.0	Steel
8-A	0.504	34,000	49,500	43,000	2	20.0	41.0	W. I.
9-A	0.501	85,700	122,000	76,000	2	23.0	63.0	Steel
10-A	0.501	46,500	65,500	47,200	2	35.0	64.5	Steel
14-A	0.407	43,850	64,500	44,600	1.62	37.1	58.5	Steel
15-D	0.439	33,100	55,200	39,700	1.75	40.5	63.0	Steel
16-A	0.376	103,500	127,000	84,700	1.5	24.0	51.7	Steel
16-FF	0.376	08,000	140,000	98,200	1.5	20.0	55.2	Steel
16-G	0.376	112,500	136,000	95,000	1.5	20.0	65.0	Steel
16-L	0.376	102,000	121,000	81,000	1.5	22.0	53.2	Steel
16-Q	0.376	110,000	131,500	95 600	1.5	21.3	54.0	Steel
16-Z	0.376	117,000	137,700	93,000	1.5	18.65	65.7	Steel
17-F	0.470	41,500	81,500	58,400	1.87	31.0	60.5	Steel
17-I	0.470	72,500	122,500	72,500	1.87	16.6	63.7	Steel
18-B	0.505	60,000	94,000	69,000	2.0	27.0	60.2	Steel
19-C	0.505	120,000	139,000	105,500	2.0	17.5	49.0	Steel
19-L	0.439	102,500	124,000	87,500	1.75	22.3	55.5	Steel
20-A	0.502	111,500	126,000	74,000	2.0	17.0	65.5	Steel
20-E	0.501	106,700	122,000	69,000	2.0	16.0	67.0	Steel
21-F	0.376	46,900	84,500	61,300	1.5	32.0	59.5	Steel
21-K	0.376	....	95,000	65,000	1.5	19.3	58.6	Steel
21-M	0.376	40,500	69,400	4[illegible]200	1.5	36.6	71 6	Steel
22-E	0.376	39,600	70,400	45,[illegible]90	1.5	37.3	70.0	Steel
22-I	0.376	39,600	69,400	44,10[illegible]	1.5	37.3	71.2	Steel
22-L	0.376	40,500	68,500	42,300	1 5	38.6	70.8	Steel
23-E	0.219	135,000	148,500	83,800	0 875	20.6	67.0	Steel
24-B	0.221	50,000	65,800	47,300	0 875	19.4	63.2	Steel
25-B	0.256	126,500	140,000	89,300	1.0	19.0	66.0	Steel
29-C	0.250	94,000	116,000	79,500	1.0	20.0	61.0	Steel
30-G	0.346	116,000	131,000	89,000	....	29.8	60.0	Steel
30-H	0.315	117,000	135,000	102,500	[illegible]5	21.6	52.2	Steel
30-L	0.314	119,500	142,000	99,000	1.25	20.0	56.0	Steel
32-A	0.313	48,000	56,000	52,000	1.25	12.0	26.0	W. I.
32-A	0.315	37,600	50,500	50,500	1.25	16.0	32.5	W. I.
32-B	0.316	38,400	42,200	60,000	1.25	35.0	66.5	Steel
32-E	0.312	30,000	52,400	44,500	1.25	28.0	47.5	W. I.
32-Q	0.250	40,800	61,200	43,800	1.00	27.0	65.4	W. I.
33-A	0.313	78,000	90,000	67,500	1.25	18.4	49.4	Steel
36-A	0.252	49,000	87,500	67,000	1.00	30.0	42.6	Steel
36-D	0.250	50,000	90,000	65,000	1.00	30.0	58.0	Steel
37-A	0.220	34,200	52,500	52,500	0.875	21.7	35.2	W. I.
37-B	0.251	42,800	55,000	49,000	1.00	21.0	32.6	W. I.
37-D	0.220	40,700	56,500	47,500	0.875	25.7	47.5	W. I.
37-H	0.221	40,700	56,600	47,400	0.875	23.0	47.0	W. I.
39-A	0.439	36,300	53,000	43,500	1.75	31.4	50.0	W. I.
39-FF	0.406	34,000	53,500	45,000	1.62	23.5	43.4	W. I.
39-G	0.439	33,800	54,300	46,400	1.75	25.1	47.6	W. I.
39-I	0.407	45,000	73,000	55,400	1.62	30.8	58.5	Steel
39-H	0.407	31,200	52,300	42,800	1.62	28.4	55.4	W. I.
39-L	0.437	25,200	41,500	35,200	1.96	25.0	56.0	W. I.
41-A	0.252	51,000	143,000	....	1.00	15.0	57.0	Steel
46-B	0.251	38,800	51,000	40,800	1.00	25.0	38.0	W. I.
46-C	0.284	44,300	63,300	47,500	1.125	22.1	9.2	W. I.
47-A	0.284	36,500	53,000	47,500	1.125	31.0	47.5	W. I.
47-B	0.283	36,500	53,000	46,000	1.125	30.0	47.5	W. I.
49-A	0.221	43,500	66,000	47,500	0.875	40.0	66.0	Steel
50-A	0.251	37,000	56,000	47,000	1.00	24.0	47.0	W. I.
50-B	0.251	39,000	55,000	49,000	1.00	28.0	34.8	W. I.
51-A	0.346	44,700	53,200	45,800	1.37	33.5	52.0	W. I.
51-B	0.345	35,500	51,600	45,200	1.37	30.6	53.6	W. I.
51-C	0.347	37,200	53,200	45,700	1.37	25.5	54.3	W. I.

NOTE: In carrying out the tensile tests reported in this table, bolts of all sizes were used. It was therefore impossible to prepare standard 0.505-in-diameter tensile specimens in all cases. Where the diameter was less than 0.505 in., the gage length was determined by the relation $L/\sqrt{(a)} = 4.5$, where L = gage length and a = cross-sectional area of test section.

The argument which is usually advanced in favor of the use of wrought iron is that it exhibits greater ductility than any type of steel. In order to check up this question, along with some others which will be mentioned later, tensile tests were made upon a series of 71 steam-line and valve bolts. Each bolt was examined microscopically to determine the material from which it had been made, and it was found that nearly a third of them had been made from wrought iron. The results of these tests are shown in Figs. 4 and 5, where it may be seen that the average ductility of the wrought-iron bolts as shown by the percentage elongation and percentage, reduction of area is about 18 per cent less than that of the mild-steel bolts, and no better than that of the high-carbon and heat-treated steel bolts which possess much greater elastic limits and tensile strengths. Table 2 contains the complete data from which Figs. 4 and 5 were prepared.

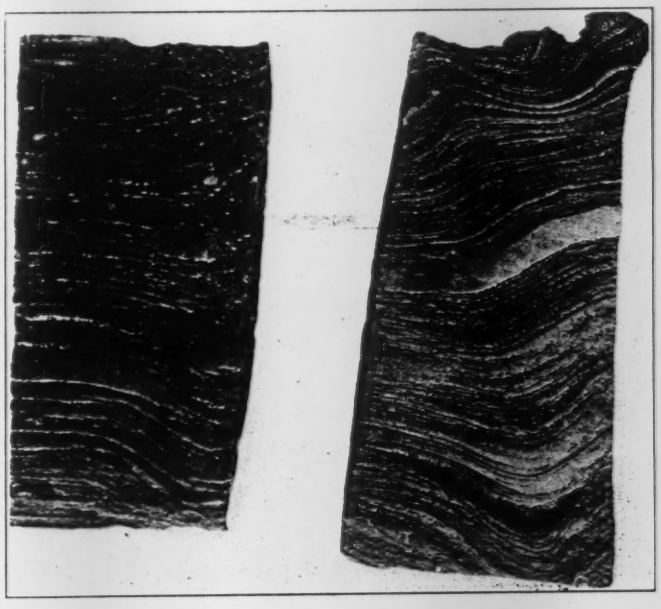


FIG. 2 LONGITUDINAL SECTIONS OF BOLTS NOS. 1 AND 2

(Etched for 1 hr. in 10 per cent sulphuric acid. The bright sections are bands of steely structure, while the dark bands are wrought iron and slag. Magnification, × 3.)

STEEL VS. SCREW STOCK

Another situation which has been discovered in this investigation is the use of screw stock as a material from which to manufacture nuts. No instances were found where screw stock had been used in making the bodies of bolts, but a great many nuts were found to have been made from this material. The author cannot help but express his conviction that the use of this material is more dangerous, because of its treacherous brittleness, than the use of wrought iron.

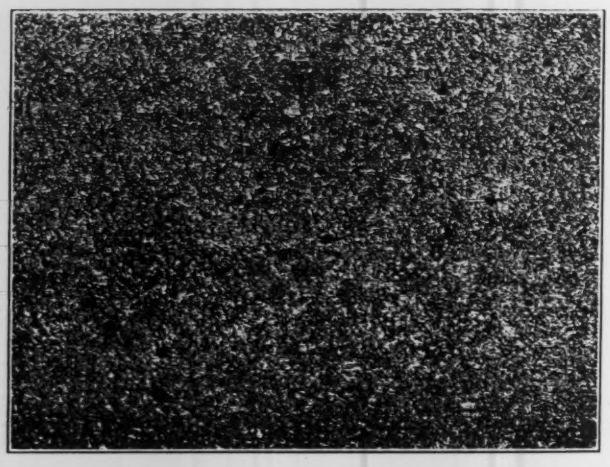
It might be well to develop this proposition through a specific example. Two nuts were submitted which had cracked while being tightened. The chemical analyses of these two nuts were as follows:

	Carbon	Sulphur	Phosphorus
Nut No. 4.....	0.16	0.093	0.108
Nut No. 5.....	0.17	0.097	0.111

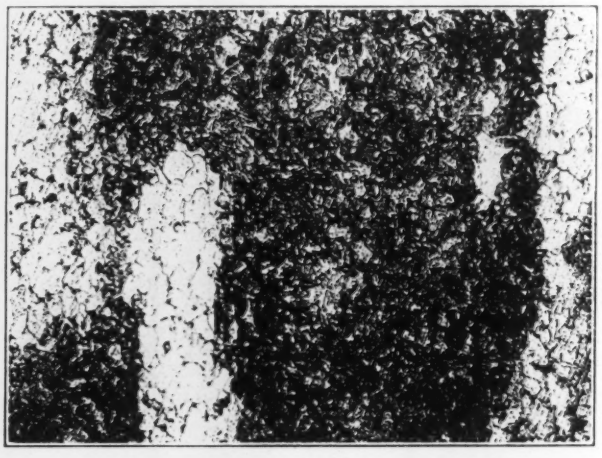
It may easily be seen from the analysis that these nuts were made from screw stock. The sulphur and phosphorus contents are just about double what they should be for freedom from brittleness. These analyses alone would condemn these nuts, but a consideration of the photomicrographs brings to light a further reason for their failure. Photomicrograph No. 1 of Fig. 6 represents an etched

longitudinal section of nut No. 5. The dark areas are pearlite, which is the carbon-carrying constituent. The light-colored crystals are free ferrite. This structure is normal for screw stock. Photomicrograph No. 2 of the same figure is an unetched longitudinal section of nut No. 5. The small dark patches at the top are slag streaks, while the large dark portions at the bottom of the section are pits or spongy spots. Such a condition represents a very poor metal even for screw stock, and it is very easy to see why these nuts cracked when a wrench was applied to them.

In a lot of bolts and nuts which was recently examined it was found that two-thirds of the nuts were made from a low-chrome steel while the other one-third were made from screw stock. Almost invariably a bolt which carried a screw-stock nut on one end carried a chrome-steel nut on the other end. Such practice is at best inconsistent. It might be termed "sending out a boy to do a man's work." Screw stock is not a satisfactory material for steam-line-bolt nuts.

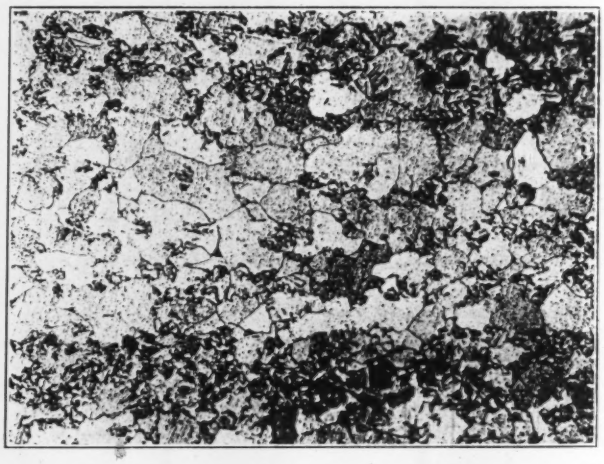


No. 1 Longitudinal section of a hardened 0.4 per cent carbon steel bolt.



No. 3 Section of nut No. 2 made from bundled scrap.

No. 2 Section of nut No. 1 made from bundled scrap.



No. 4 Section of nut No. 3 made from bundled scrap.

FIG. 3 PHOTOMICROGRAPHS OF SECTIONS OF A BOLT SHANK AND OF THREE NUTS
(Etch, 4 per cent picric-nitric acid mixture; magnification, × 100.)

THE BEST CARBON CONTENT OF BOLTS

It is well known that one of the first requisites of good bolting is a sufficient amount of ductility. This is just as important as the possession of sufficient strength. As has been pointed out, it is probably on account of a supposed superiority in this respect that wrought iron is used so much as bolting. When, however, a bolting material fails from lack of strength, there is danger of going to the other extreme and increasing strength at the expense of ductility. Reference to Fig. 5 illustrates this point. The percentage elongation of the high-carbon and heat-treated bolts is about the same as that of wrought iron (Fig. 4), while the reduction of area averages slightly less than that of the mild-steel samples. While the high-strength bolts of Fig. 5 are entirely satisfactory, they represent the tendency mentioned, and were the analyses and heat treatment varied in such a way that the tensile strength would be increased notably, a point would be reached at last where ductility would drop to a dangerously low point. Lack of ductility, of course, means brittleness.

It is a natural conclusion from what has just been said that the best carbon content for bolting steel lies somewhere between 0.10 per cent and 0.6 per cent. It would seem that the range from about 0.20 per cent to 0.45 per cent would yield the best combination of tensile strength and ductility in either the heat-treated or unheat-treated condition.

STANDARD PROCEDURE A NECESSITY

It would be possible to multiply the foregoing examples many times, but the result would be only to emphasize what is already evident, namely, the lack of any uniform and consistent procedure in regard to the type of bolts which are used in the construction of steam-power-plant equipment. It is felt that it is unnecessary to argue in favor of the adoption of some standard procedure, for the thing seems obvious. One power plant, recognizing the necessity for covering this material with a rigid specification, has drawn up and adopted a specification under which they have been buying and inspecting bolts. It will not be included in this paper, for it is proposed to present for consideration of the Society the tentative specification for high-temperature-alloy bolting which has been prepared by Committee A-1-22 of the American Society for Testing Materials. This specification is here included in full.

TENTATIVE SPECIFICATION FOR HIGH-TEMPERATURE-ALLOY BOLTING

1 *Material Covered.* This specification covers bolting material for valves, flanges, fittings, and similar parts for temperatures up to 750 deg. fahr.

I—MANUFACTURE

2 *Process.* The steel shall be made by one or more of the following processes: open hearth, electric furnace, or crucible.

3 *Discard.* A sufficient discard shall be made from each ingot to secure freedom from injurious piping and undue segregation.

4 *Heat Treatment.* (*a*) Heat treatment shall consist of quenching and tempering.

(*b*) The procedure to be followed in quenching and tempering shall consist in allowing the objects, immediately after rolling or forging, to cool to a temperature below the critical range, under suitable conditions, to prevent injury by too rapid cooling. Each lot (lot representing materials of essentially the same composition) shall then be uniformly heat treated to the proper temperature to refine the grain (a group thus reheated being known as

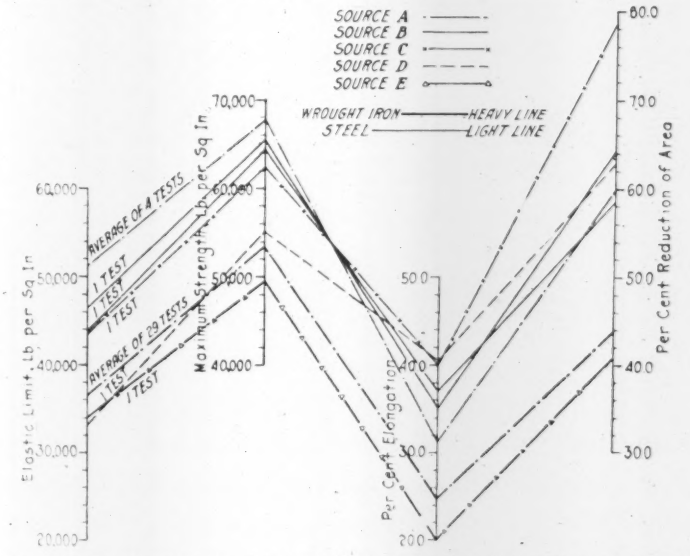


FIG. 4 COMPARISON OF TENSILE PROPERTIES OF MILD-CARBON-STEEL AND WROUGHT-IRON BOLTS

a "quenching charge"), and quenched in some medium under substantially uniform conditions for each quenching charge. Finally, each lot shall be uniformly reheated to the proper temperature for tempering or "drawing back" (a group thus reheated being known as a "tempering charge"), and allowed to cool uniformly.

II—CHEMICAL PROPERTIES AND TESTS

5 *Chemical Composition.* (*a*) The following steels having a content of from 0.20 to 0.45 carbon are considered suitable for this service: namely, nickel, low-chrome nickel, chrome-vanadium, and chrome-manganese. Other alloy steels or compositions may be used subject to the approval of the ultimate purchaser.

(*b*) *Approval.* The manufacturer shall submit to the purchaser for his approval the chemical composition of the material which he proposes to furnish.

(*c*) All steels so purchased shall conform to the following requirements as to chemical composition:

Phosphorus, per cent	not over 0.04
Sulphur, per cent	not over 0.05

6 *Ladle Analyses.* An analysis of each melt of steel shall be made by the steel manufacturer to determine the percentages of the elements present. This analysis shall be made from a test ingot taken during the pouring of the melt. The chemical composition thus determined shall be reported to the purchaser or his representative, and shall conform to the requirements in Section 5.

7 *Check Analysis.* Check analysis may be made by any purchaser of billets, bars, rods, bolts, studs, or other form of bolting material from drillings from the above, or from test pieces representing them. Such purchased materials must be capable of proper identification, and analysis shall be made within a reasonable time after receipt of the shipment if the vendor is to be held fully accountable. Check analysis shall conform to the requirements of specification in Section 5, except that the rejection limits for phosphorus shall be 0.050 per cent, and for sulphur, 0.055 per cent.

III—PHYSICAL PROPERTIES AND TESTS

8 *Tension Tests.* (*a*) The bolting material shall conform to the following tensile properties:

	A	B	C
Ultimate strength	95,000–115,000	105,000–125,000	125,000 min.
Yield point	70,000 min.	80,000 min.	105,000 min.
Elongation in 2 in.	20% min.	20% min.	16% min.
Reduction of area	50% min.	50% min.	50% min.
Brinell hardness nos.	190–250	210–270	260–320

(*b*) Brinell hardness tests shall not be used as a basis for rejection, but shall indicate that confirming tensile tests shall be made.

(*c*) The yield point shall be determined by the drop of the beam of the testing machine, or by physical measurement, and shall conform to the requirements specified in Section 8 (*a*).

(*d*) Physical tests of bolting material shall be made only after final heat treatment.

9 *Tension-Test Specimens.* Tension-test specimens shall be taken from either the bars or bolts.

10 *Number of Tests.* (*a*) One tension test shall be made from each lot of 100 pieces or less, from each quench or "drawback" charge of either bars or bolts. If more than one melt is represented in any quenching charge or temperature charge, a test must be made from each melt.

(*b*) If any test specimen shows defective machining or develops flaws, it may be discarded and another specimen substituted.

(*c*) If the percentage of elongation of any test specimen is less than that specified in Section 8 (*a*), and any part of the fracture is more than $^3/_4$ in. from the center of the gage length as indicated by the scribe scratches marked on the specimen before testing, a retest shall be allowed.

(*d*) In case of small sections which will not permit of taking the standard tension-test specimen, then the size of the tension-test specimen to be used will be in accordance with Section 5 of E-8-24T, Tentative Methods of Testing Metals or Metallic Substances.

(*e*) Certification that the material meets these specifications shall be made by the manufacturer to the purchaser, and such certificates for bolting materials for valves and other fittings intended for stock or other purposes requiring assembly in the manufacturers shop shall be accepted in lieu of the above tests unless otherwise specified on the purchase order; and that Section 10(*a*) shall not apply on purchase orders for one hundred (100) pieces or less, unless specifically required by purchase order.

11 *Retests.* If the results of the physical tests of any lot of bolts or bolting material do not conform to the requirements specified, the manufacturer may retreat such lots one or more times and retests shall be made as specified in Section 10(*a*).

IV—WORKMANSHIP AND FINISH

12 *Workmanship.* Bolt studs shall be threaded at both ends and equipped with cold-punched semi-finished carbon-steel nuts of U. S. Standard rough dimensions, chamfered and trimmed. If washers are used under

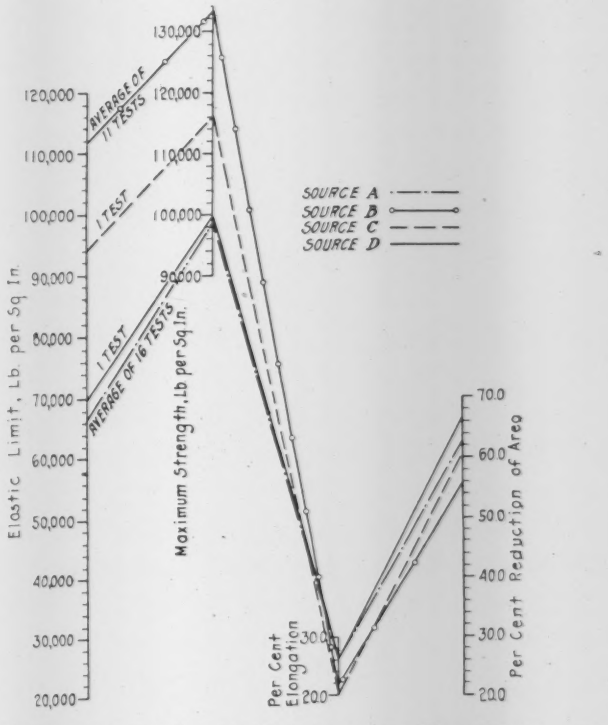


FIG. 5 COMPARISON OF TENSILE PROPERTIES OF HIGH-CARBON AND HEAT-TREATED STEELS

nuts they shall be of forged or rolled steel. All nuts and washers shall be free from injurious defects and shall have workmanlike finish, and conform to chemical requirements with respect to sulphur and phosphorus as indicated in Sections 5(*c*) and 7.

13 *Threads.* U. S. Standard threads shall be used, except that no bolt stud shall have less than eight threads per inch.

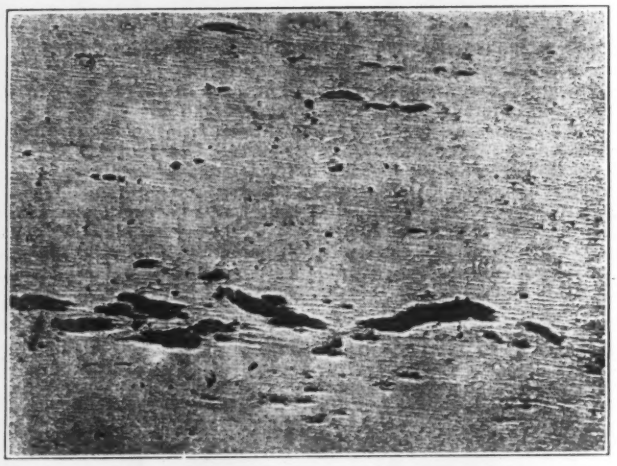
V—MARKING

14 *Identification Marks.* Identification marks shall be legibly stamped on each charge of bolt studs or bolting material. The manufacturer shall indicate the location of such identification marks.

VI—INSPECTION AND REJECTION

15 *Inspection.* (*a*) The inspector representing the purchaser shall have free entry, at all times while work on the contract of the purchaser is being performed, to all parts of the manufacturer's works which concern the manu-

No. 1 Etched section, showing dark crystals of pearlite in a mass of ferrite. Note banded structure, which is indicative of poor strength.



No. 2 Unetched section showing slag and pits.

FIG. 6 PHOTOMICROGRAPHS OF LONGITUDINAL SECTIONS OF BOLT No. 5
(Etch, 4 per cent picric-nitric acid mixture; magnification, × 100.)

facture of the bolt studs ordered, and the manufacturer shall afford the inspector, free of cost, all reasonable facilities to satisfy him that the bolt studs are being furnished in accordance with these specifications. Tests and inspection at the place of manufacture shall be made prior to shipment.

(*b*) The purchaser may make the tests to govern the acceptance or rejection of the bolting material in his own laboratory or elsewhere. Such tests, however, shall be made at the expense of the purchaser.

(*c*) Tests and inspection shall be so conducted as not to interfere with the operation of the works.

16 *Rejection.* (*a*) Unless otherwise specified, any rejection based on tests shall be reported within five working days from the receipt of the samples.

(*b*) Bolting material which shows injurious defects while being finished shall be rejected.

17 *Rehearing.* Samples tested in accordance with Section 7 which represent rejected bolt studs shall be preserved for two weeks from the date of transmission of the test report. In case of dissatisfaction with the results of the test, the manufacturer may make claim for a rehearing within that time.

The rigid adoption of this specification will eliminate all the objectionable conditions and materials which have been referred to earlier in this paper. Sections 2, 5, and 12 eliminate the possibility of using wrought iron or screw stock as material for either bolts or nuts, and also limit the carbon content of the steel used to a range where ductility will not be too low on finished bolts.

It is mentioned as a final thought that, while it may be perfectly satisfactory to use high-quality carbon-steel bolting on the same job with alloy-steel bolting, there is always the danger of mixing up the two types; and since the difference in price is quite small, it is recommended that alloy bolting be used on all power-plant work.

The Present State of Industrial Psychology

A Consideration of the Science and Art of Handling Men, and an Evaluation of the Successes and Failures That Are Found

By LILLIAN M. GILBRETH,[1] MONTCLAIR, N. J.

INDUSTRIAL psychology is accepted today both by engineers and psychologists as useful and valuable. This is a distinct advance over conditions ten years ago. A review of its rise in prestige justifies a survey of present conditions.

Industrial psychology is the study of men as they behave under industrial conditions. This paper will confine itself to a consideration of the science and the art of handling men as it is seen in industry today, and to an evaluation of the successes and the failures that are found.

For purpose of discussion, we shall divide the subject into four parts:

1 Selecting or finding the right job for the right man
2 Teaching or training the man for the job, for promotion and for satisfactory adjustment to industrial life
3 Securing and directing technical coöperation, or training men to work together in production activities
4 Directing group contacts or training men to function as group members and groups to function together.

Theory and Technic in Selection

In the field of selection we find today not only well-thought-out theories as to efficient selection, but a gradually developing technic. Selection is being done increasingly by an employment manager or personnel man or by an employment department, rather than through the older policy of having any department that requires a worker interview the applicant and make the selection for itself. This is in line with the tendency toward functionalization and specialization which has been growing since the advent of Scientific Management.

An employment department must know the general type of worker that the organization desires. There must also be an intimate knowledge of the various types of work done; of the qualifications necessary to fill the various positions in the organization; of both the opportunities and the limitations of each job; of the type of training afforded; of the type of promotion available; of the worker's surroundings, equipment and tools, and of the work method used. These are necessary to present a satisfactory outline of the work to the possible worker and to evaluate him in view of the total situation that he is to face. There must be a knowledge of the science of selecting men, and of that art of selection which includes gaining the applicant's confidence, noting and evaluating his behavior and attitude as well as his specific responses, and being able to put one's self in his place in order to estimate correctly both what he says and what he means.

The advantages of having the employment department do the selecting instead of the department where the man is to be used are the possibility of centralizing there the technic of selection and delegating it to trained selectors. These have a bird's-eye view of the needs and opportunities of the whole organization and are thus able to place a desirable applicant to better advantage.

There are, however, certain advantages in having the department which is to use the man do the selecting. These advantages of selecting by the working force and in the work place can be partially transferred to the employment department by sending a member of this department with the applicant to the work place before the final hiring is done. He can note the impression that the job and work place make upon the applicant and that the applicant makes upon the department. He then gets an idea of possible adjustment and often prevents a later departure, dismissal, or transfer.

There is no better test of acquaintance or intimate knowledge of an occupation than seeing the worker in the proposed work place. The way that he looks about is indicative. The comments that he makes are significant, and if he once handles the machinery or tools, his experience and skill can be fairly estimated.[1]

The Questionnaire Method

There are both advantages and disadvantages to the questionnaire method of selection. Success presupposes that the applicant understands clearly *why* the questions to be answered are asked and exactly *how* he can profit if they are answered correctly and fully. It must be made clear that it is the aim of the organization to know everything possible about its workers, not only to find the jobs for which they are best fitted but to give opportunities for overcoming handicaps and developing abilities, and to provide for constant promotion.

Extended descriptions of the types of work done by the applicant in previous jobs are useful in determining his suitability for the kind of work offered. Information as to what books have been read, what subjects are of interest, ambitions, likes and dislikes, and other emotional reactions, is of great value in forecasting achievement.

It is often best to offer to fill in an applicant's blank for him if he hesitates to do it—as many a good worker hesitates to express himself at length in writing—but this matter must be handled carefully, as some types resent assistance. Sometimes a valuable worker is of an inarticulate or suspicious nature. It may then be wise to postpone any detailed questions to a later period, but any material gained during the selecting period helps to better selection.

Tests

The value of tests in selection has been debated from every possible angle. Information tests are of use for positions where it is essential that the applicant have certain facts at his disposal. Intelligence tests, at first so overrated, have of late been much underrated but will prove of increasing value. At present the soundest and most conservative investigators in this field claim only to have standardized such tests as determine the lower boundaries of intelligence. It is possible to locate such applicants as are unfit for the industry to which they are applying; while the requirements remain what they are, many jobs are now being filled by those of far too high a type of mentality. We may expect in the future more rather than fewer morons to be employed in industry. Better teaching in industry makes possible selecting workers of a lower type who can perform the work if they are developed to the height of their ability, and who will receive permanent satisfaction both in the performing and in the wages received. This involves retraining and promotion of the more highly gifted.

No sound psychologists claim today that intelligence tests can determine the upper boundary of intelligence. It is therefore never safe to suppose that one knows exactly to what heights of promotion an applicant can be trained, or to make recommendations which restrict training and promotion.

Satisfactory tests of specific abilities cover today only tests for such simple things as reaction time, and it is not to be expected that an applicant for any complicated type of work can be thoroughly evaluated by such tests as can be given during the usual test period. One reason, however, why such tests have not proved more satisfactory is because industry has not furnished the proper criteria and records which the psychologist must have in order to prove the correctness of his findings and to shape his technic in accordance with his results. Industry is also, at least partly, to blame that there has not been a closer coöperation between those best acquainted with the work and those best versed in the technic of giving tests. There is frequently little understanding as to which abilities are necessary for any special type of work. Some excellent tests have been devised by those in industry who know the technic

[1] Frank B. Gilbreth, Inc., Consulting Engineers.

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[1] Skill in Production, by L. A. Legros. Presidential address, British Section, Société des Ingénieurs de France, 1925.

Be sure to bring this copy with you to the Annual Meeting

of tests and have studied their operations and reduced them to such form that they can be tested in small, well-defined units in a laboratory. We believe an increase in this type of successful testing is to be expected. Such tests are best given in the plants themselves and not in outside psychological laboratories, as it is practically impossible to duplicate working conditions or working incentives in the latter. Research work should be carried out in a laboratory outside the plant, where variables can be controlled, *but applications to industry must be made in industry.*

Trade tests are successful where the applicant is actually tried out on *essential parts* of the specific trade to which he is to be put, either in the work place itself or in a plant laboratory which parallels working conditions.

Psychological tests should be accompanied by a physical examination and also by a psychiatric examination as extended as is practicable. One of the most optimistic psychiatrists at present in industry feels that a successful test demands a day for each subject, while an analysis may take a year or more. Doubtless this period can be shortened, as material accumulates through the records of the various cases and the classification and correlation of the data. Where a psychiatric examination is not available, the psychologist or the employment director can do much by recording the behavior of subjects during the hiring period and noting especially emotional reactions. The results of all tests should come finally into the hands of some person qualified to combine them and judge them as a complete picture of the personality being examined.

Sex and Selection

There is much argument in industry today as to whether selection of workers is a job best filled by a man or a woman. Some industries seem to feel that men have greater aptitude for the job of selecting, while still others believe that the opposite is true. Others feel that a man is a better judge of the capabilities of men, has a better understanding of the technic of various jobs, is less apt to be deceived and more apt to judge dispassionately. The same type of reasoning holds where women are in charge of the hiring of women. There seems to be no special reason why an industry should not prefer to have a man hire the women, and a woman hire the men. A complete survey of existing practice in this field is lacking. It should show *who* is doing the hiring, the number of men and women who are being hired, methods used, reasons, and results.

Selection for Study, Transfer and Promotion

Selection for other work of employees who are already hired and at work is a simpler problem, in that considerably more is known of the applicant for special work than is known of the applicant new to the industry or the job. If workers are being selected to be motion-studied, it is customary to choose the best available at the work to be investigated. This is determined, first, by a study of production and other records available; second, by the opinions of the group of workers involved, as to relative ability; third, by the opinion of the supervisors; and finally, as the result of observations, by those most expert in judging efficient methods. To the group selected for study is invariably added the laziest worker, who may show efficient methods or motions which enable him to gain a satisfactory record with the least expenditure of effort and the smallest amount of fatigue.[1]

There are various accepted methods of selecting workers for promotion. A carefully perfected plan is the so-called "three-position plan of promotion," which provides that every worker occupies constantly three positions in the organization—that of teacher in the position that he has left, that of worker in the position that he occupies, and that of student in the position he desires to occupy.[2] Where this is in use throughout an organization, selection for promotion becomes easy.

Relation of Selection to Training and Coöperation

Selection is closely related to the other divisions into which this consideration of the subject of handling men in industry has been divided. Proper selection makes easier training. A certain amount of teaching can be done during the selection period by making the worker acquainted with the organization and work, and some selection can be done during the training period, if those who have through some oversight been mistakenly selected are dropped or fitted to other work before a serious maladjustment has taken place.

The psychology of good will which influences the directing of group contacts has much to do with emotional estimates. Here also much can be done during the period of first selection to note indications along these lines that will make selection of those who shall function in this type of coöperation easy.

Problems of Training

It is a distinct advance in the application of psychology to industry that there is an increasing appreciation of the fact that every worker in industry must be constantly taught if he is to attain his greatest efficiency. Training in the work of the organization is usually given in the plant itself, either at the work place or in the training department, by some one who has come up through the plant work and become an instructor, or by some one who has been a teacher and who undertakes the training after having gained some experience of the plant work. If the worker has had some vocational guidance before coming into industry and has been carefully selected, the training can be most efficiently undertaken.

While the new worker who has had no industrial experience furnishes the need for the most prolonged training, the problem is simple because no bad industrial habits are likely to have been formed. If the teaching is efficient the worker acquires the right habits quickly. While the period of training, either in the training department or the operating department, before work is actually undertaken demands considerable of the instructor's time, there is not so much need of follow-up for fear of habit lapses.

The worker who has had industrial experience and is new only to the special plant may not require as much training in work methods but may need more emphasis on training in *right* habits and a careful follow-up to see that there is not a lapse into old habits, with the consequent decrease in efficiency.

The worker who is experienced both in industry and in the plant furnishes a special problem in training. Even if there is apparently no change in method, his work should be inspected periodically from the method standpoint. Any lapses should be corrected and the instructor should add new information in theory or technic of the work. If a new method is introduced the training problem is important. The work habits of this type of worker are deeply fixed. He must understand not only the *what* and the *how* of the changes, but the *why* as well, and he must be furnished an incentive not only to carry him through the period of learning, but through the period of use of the new methods, until they become as automatic as the old ones they supplant. The psychology of the total situation demands a consideration of the emotional reactions as well as of the intellectual processes involved during this training.

Training may concern itself with an old worker at a new job. This may be because of transfer on account of poor, slack, or good work, or because of promotion. It is essential to discover the *likenesses* between the habits of work used in the older type of work and in the newer. Any transferable units should be utilized and a proper learning process of the new units outlined. In many types of industry today, old workers are being taught new jobs and new workers are being taught several jobs in order that they may be transferred from one to another if work becomes slack, or in order that seasonal depressions may be avoided. Such plans should consider likenesses and through them make the worker efficient in teaching the man who is to follow him, as well as learning the work of the man ahead of him.

Classifications for Training

The workers who are trained may be classified as normal, subnormal, and supernormal, thinking of them chiefly from the mental standpoint. Subnormals of various types have been taught to become profitable members of the industrial community. The teaching of the subnormal outside of industry has sometimes seemed to be out of proportion to that of the normal and supernormal. But the fact that subnormals are prone to certain crimes and that training makes them not only less susceptible but often useful, justifies careful teaching. Subnormals can be taught to perform satisfactorily tasks in industry which are unpleasant or monotonous or lacking in incentive to a normal worker. It should be understood that the aim of training of all subnormals is to make them as nearly

[1] Psychology of Management, Chap. 4. [2] Applied Motion Study, p. 197.

normal as possible. The chief interest should always center on the normal and supernormal, who are the strength and leadership of the working world.

The best opportunity for standardization lies in the field of the normal, who may be expected to respond in a predictable fashion to interests and learning processes. The training of the supernormal is facilitated by the functionalization and specialization supplied by Scientific Management,[1] but it is necessary to note that while society can receive great benefits from the intensive training of the supernormal, there is danger of dissatisfaction if the individualistic side is overstressed and if differences are cultivated to the neglect of fundamental likenesses.

With standardization in view, workers may be classified into the normal and the abnormal. The aim of abnormal training is to bring the abnormal as near to the normal as possible. Certain types of occupations which develop abnormality in individuals who may have only latent tendencies, may become safety valves or outlets of energy for other types who demand such activities as these occupations offer.

Workers may be classified from the learning standpoint as those who learn with extreme rapidity, those who learn easily, those who learn with slight effort, those who learn with great effort, and those who have practically stopped learning. The last is one of the most hopeless types to be found, but having stopped learning does not always mean having lost the capacity to teach. Their surroundings, equipments, work methods, and tools may be bettered, and they may be allowed to stay at the type of work that satisfies them on condition that they teach others, who must learn it as a part of their preparation for similar work or for promotion.

From the skill standpoint, workers may be classified as expert, efficient, inefficient, passable, poor or failures; or as dexterous or fumbling; or as fully equipped for the work or handicapped. There are as many classifications as there are viewpoints. The psychology of the total situation demands that as many of these viewpoints as practicable be considered, not forgetting the emotional standpoint, and that there be some classification of the work and workers according to the emotional demand.

Evaluating Jobs

Applied psychology has devoted much attention to the field of matching the worker to the work. Every job in a systematized or scientifically managed organization is analyzed, and standard job specifications are a part of orthodox present practice. The first step in this process is to make a survey as to what the job covers *as it is done*. This is supplemented by an evaluation of the job. The psychiatrist studies and evaluates the job from the emotional standpoint. He tells for whom such work would be bad, whom it would not affect, whom it would help. The psychologist states the mental ability that it demands. The physiologist estimates the amount and kinds of energy involved and the fatigue. The physician, the planning department, the employment department, the training department, the supervisors or department managers and efficient workers who have had experience with it and are able to say which things help in doing the work and which hinder, what one may expect from the work as to production and satisfaction as well as the possible evil effects—all add their evaluations. The combination of these findings gives data which form a picture of the total situation of the worker at the work.

When a complete evaluation of a job has been made, all records should be consulted to verify the evaluation. Turnover records, records which would show what types of workers occupied the job, for how long and how successfully, waste reports, and incentive reports will prove of value. Among these should be mentioned specially motion-study records. The "therbligs," or elements of a motion involved in doing the work, are listed and evaluated as results of the application of micromotion study.[2] These furnish an admirable check on evaluation. They also point out likenesses to other jobs and give information for selection of new workers, for methods of teaching workers, and for promotion.

The motion-study department furnishes data useful to all departments that have to do with the human element, and methods of measurement for the activities of these departments. It correlates the work of the planning department, training department, employment department, psychologist, psychiatrist, and other investigators of the human element in industry.

Maintenance of Teaching Efficiency

Teaching is not complete when the method has been standardized and the group has been taught to that point where its members can perform the activity to their satisfaction and it has become such a habit pattern that there is little fear of lapse into an older, poorer method of performing the work. At stated intervals the job should be resurveyed. Automaticity, "the greatest free asset of the workingman," is the result of habit. It is best attained by following the well-established laws of habit formation, so excellently stated by James[1] and applied in industry by Hartness[2] and others.

The habit should be started with great impetus, repeated as often as possible, and under the most careful supervision, in order that no lapses shall occur. Habits never become so absolutely assured that one can attempt to omit the inspection or maintenance factors. At certain set times, which vary according to the ease with which the habit may be changed, the habits of the worker at the work should be carefully surveyed. A comparative survey will not only indicate errors which should be corrected, but better methods which should be standardized. It is not necessary to indicate in detail where, when, how and why teaching is best done, according to the laws of advanced practice.[3] Teaching should be done both in the training department and in the work place itself, in order that positive, established habits may carry over under working conditions and not lapse when new variables enter the situation.

In the teaching field there have been contributions both from education to industry and from industry to education. Psychology is equally applicable to both fields, and has made distinct advances in each which can be transferred to the other. Motion study, fatigue study and skill study—all are industrial outgrowths of the correlation of psychology with management which can well be transferred to the educational field. When fundamental likenesses between educational problems *out* of industry and *in* industry are recognized, we may expect analysis into transferable units and for actual transfer. The most rapid and proficient advancement can be made when every industrial plant faces both its research and its transfer problems itself, and develops its own type of training, being careful to utilize all existing activities and to develop only from that point on, where these outside activities cease to be of service. Safety, health, posture, fatigue, monotony, heat, light, ventilation, and many other subjects are already being thoroughly investigated by specialized bodies. Data should be studied and utilized before plant investigations are made. There is a sufficient field in adapting such data to the use of the plant, and in making such investigations only as the peculiarities of the plant problems demand.

Standardizing Technical Coöperation

A well-taught worker can easily be directed in technical coöperation. The research work done in the industry also assists here.

Since coöperation depends primarily on a knowledge of the exact requirements of the work and of the methods by which it can best be done, every job in the organization may be considered as related to every other job through its likenesses and differences. Jobs are related also through the relationship of the people who are working at them. For example, the job of the instructor and the job of the learner, when both are workers; the job of the worker and the job of the helper, at such work as bricklaying, where the helper brings the material and deposits it in such a way that it is most easily available for the worker, or in such a job as the braider assembly or the assembly of the many parts of a motor, where the helper puts the parts needed upon the packet and the worker takes them from there in the shortest sequence and assembles them with the least amount of fatigue.[4] Jobs are related as the worker is related to his foreman, his inspector, his supervisor, the planning department, the training department, etc. Jobs are most closely related when they stand in a dependent sequence, when each succeeding worker in a process is dependent upon those who precede him for the opportunity to do his work.

[1] Psychology of Management, Chap. 3.

[2] *Bulletin* of the Taylor Society, June, 1921.

[1] Psychology, by William James, Chap. 4.

[2] The Human Factor in Works Management, Chap. 1.

[3] Training Employees in Production Work, *Journal*, Society of Automotive Engineers, October, 1925.

[4] Ten Years of Scientific Management, John G. Aldrich, *Management Engineering*, Feb., 1923.

Technical coöperation requires not only a knowledge on the part of those concerned in the work as to exactly *what* coöperation is necessary, *why* it is necessary, and *how* it can best be secured, but also a continued incentive and maintenance. The incentive can profitably take the form of an interrelated bonus, that is, one that makes the bonus of the man in charge depend on all those under him making their bonus; of the three-position plan of promotion or of a new type of group bonus which is being installed in some pioneer organizations. This group bonus differs very radically from the old-fashioned type of bonus—which simply meant that a certain amount was divided among all of the workers equally—in that it considers the psychology of the situation and also estimates clearly the amount of effort and time put in by each worker, evaluates these, and distributes the pay in accordance with various factors of endeavor.

Technical coöperation concerns itself primarily with intelligence and skill. This should interest especially The American Society of Mechanical Engineers, to which all the pioneers in the field of Scientific Management belonged, and which early recognized the importance of transferring skill and the conservation of skill as integral parts of industrial progress.[1] Investigations in the field of skill are being pushed by engineers and psychologists alike, both in the field of industry and in that of the sports.[2] Skill study is supplementing motion study and fatigue study, and the three together are operating both to find the elements governing technical coöperation and to provide for teaching these.

Evaluating Group Coöperation

The securing and directing of technical coöperation is largely a matter of individual contacts, worker with worker, worker with teacher, worker with supervisor, etc., and of coöperative transfer of skill. The securing of profitable group contacts has less to do with intelligence and skill than with the psychology of securing good will. Here the emotions must be considered. We have special need here for the psychology of the total situation. There must be as little emphasis as possible on the differences between the management and the men. There must be a stress on likenesses and an attempt to think of all members of the organization not as individuals but as primarily members of an industrial group.

Group contacts may be only occasional and because of some special event. Such are "drives," when the members of the organization meet to face a common problem and outline a common solution. Work is estimated in terms of the group, and divided so that each member may do what is both for his own advancement and for the group needs. It should be emphasized that, as is also the case with selection, with teaching, and with technical coöperation, the aim of such contacts may be either good or bad. Psychology is not ethics and must not be judged by ethical standards.

Group contacts may be less occasional. They may become periodic, with no set program, or periodic with a set program which may be organized by a special group—for example, the management—or by a special group—for example, the workers.

There is an increasing tendency for such contacts to take place in organized form. This can be noted in industries of all types, both where the line of demarcation between the management and the men has always been well defined, and also where it has been less sharply defined. The management may be theoretically unorganized, except in committee groups or in functional organizations which are a part of the mechanism of putting through plant business, or may be very thoroughly organized into a management group. The men may be organized into shop committees or other groups, with a gradually increasing progression into the field of self-government. It is interesting from the psychological standpoint to note that results are not always as expected. For example, the management, far from feeling that certain of its privileges are being taken from it, is usually very well satisfied to give up plant discipline and similar functions, while the working group is not as happy as it expected to be in having to take over these difficult and sometimes disagreeable responsibilities. There is often a much better understanding between managers and men when the first group see their responsibilities lessened and the second their responsibilities increased.

Even when older and perhaps antagonistic types of organizations exist, when the workers are unionized either within the plant only or in a national or international union, and the managers belong to employers' unions, a careful consideration of problems of securing good will results in transforming the activities of such associations, yet conserving their good features. No type of organization, no matter how faulty it may seem, should be done away with without considering its possible good points and the availability of much of its mechanism as a channel for improved activity. An emotion thoroughly understood is often half handled. A new contribution in this field of the psychology of good will is the analysis of emotional situations arising through thwarting instincts or allowing them too free rein. The situation is surveyed to trace the cause and a possible and satisfactory solution of the problem involved is outlined. A complete handling of this type of contact means the consideration not only of individual but of group psychology; that is, a study of individuals as they react in certain ways under certain conditions. Here again the psychology of the total situation is of inestimable value. Successful technical and successful group contacts evidence themselves in a satisfactory industrial situation, as measured by the usual production and other plant records.

It must be noted that, while the subject of handling men has been divided into four parts, this is simply an arbitrary division for purposes of discussion. A valuable survey of the efficiency of existing conditions can be made, comparing the results of each type of work; that is, of the work that is done under each of these divisions as it affects the others. For example, if the selecting is so rigidly standardized or functionalized that group coöperation and group contacts are not conserved or improved, there is room for improvement here. If the teaching has not in mind providing material for better selection theory and technic and also more efficient use of the two types of contacts, this also requires improvement. If the technical contacts do not contribute to good will, do not help the instructors and make the learning process easier as well as the process of selection, they are not satisfactory. Finally, in case the good will does not coöperate with the technic in obtaining technical coöperation and function through the selecting and training process, it also is lessened.

Coördinating Psychology and Engineering

There is perhaps no finer, more stimulating, more profitable field for investigation in psychology than industry. Such investigations are not a burden on industry, do not add new problems, or complicate existing practice. The problems have always existed, the practices recommended are simpler and more easily understood than those already in use, and as a result, take from, rather than add to, industrial burdens.

In order that the most benefits may accrue to industry from the work of the psychologist, and that the problems may be solved with least waste, the relation of the psychologist to the engineer—who is so important in industry—should be standardized. Neither engineer nor psychologist can expect to get the greatest results by working alone. Both viewpoints and both activities are needed for the most efficient solution of industrial problems involving the human element. It has been customary in this country for the psychologist to work under the direction of the engineer, while abroad it is usual for the psychologist to be in control, even when psychological and engineering findings are correlated. Excellent results have been secured by both types of coöperation, but because engineering is a far more advanced science than is psychology and because the engineer has been trained to methods of accurate measurement and is accustomed to abide by the results of such measurement, because the engineer is accustomed to recording facts at their face value, refusing to advance beyond what the facts justify, it seems best, in this country at least, for the engineer to continue in charge of the investigations and the correlations.

This does not mean that the engineer should be allowed to limit the investigations or to refuse to follow either the research or the technic of the psychologist in his peculiar field. It means only, that all industrial findings must be subject to evaluation from the engineer's standpoint and must show their value ultimately in such terms as production, for which industry primarily exists.

[1] The Present State of the Art of Industrial Management, Trans. A.S.M.E., vol. 34, p. 1131; Ten Years Progress in Management, L. P. Alford. Trans, A.S.M.E., vol. 44, p. 1243.

[2] Skill in Work and Play, T. H. Pear.

Carbon Dioxide as an Index of Fatigue

By WALTER N. POLAKOV,[1] NEW YORK, N. Y.

The problem of industrial fatigue has been approached from two sides: the effect of fatigue on work, and the effect of fatigue on the workman. Since an individual is inseparable from his environment, neither of these detached studies yielded practically useful results. In this paper the author suggests a simple and practical procedure for reducing (if not eliminating) the ill-effects of fatigue through a joint study of work and workman. Apparently immediate and reliable indications offered by the measurement of the CO_2 content in the exhalations of the workman will enable the management to organize the work so that the organism of the workman will produce a maximum result with a minimum exertion.

INDUSTRIAL fatigue is recognized as the cause of: (*a*) Reduction of quantity and quality of output, and (*b*) reduction of strength and vitality of workers. The first part of the problem has been studied since Taylor's time studies indicated the necessity of rest periods for the maximum output. The second part has received a great deal of attention abroad and at home, particularly during the war, when conservation of labor became an economic necessity.

The time studies analyzing jobs into their elements and determining the allotted time for the performace of each led to two important consequences:

a Readjustment of job elements for speeding up the operation[2]

b Interruption of work by rest periods.[3]

Further refinements were introduced by Jules Amar, Morey, and the Gilbreths by the employment of chronocyclographs and motion pictures for micromotion studies. These efforts, aiming at an increase of the net energy output of human workers, led to the elimination of unnecessary motions in the cycle of work motions, and thus indirectly reduced fatigue.

Inasmuch as the aim of all these methods was the elimination of useless motions in order to conserve energy for useful motions, the question as to which of these are fatigue-producing remained unstudied.

Another line of research was accordingly undertaken to determine the physiological factors of fatigue, irrespective of the quantity and quality of the industrial output of workers. Laboratory experiments on the fatigue of single muscles, group of muscles, and nerve centers by means of dynamometers, ergographs, etc., while shedding much light on the problem, did not take into due consideration the fact that in shop work the character of the operations usually provides cyclic relaxation. Neither did it study the worker as a whole, and, studying elements, it had little to contribute to the solution of actual problems.

The study of work from the point of view of energetics has aimed at an understanding of the nature of the conversion of energy supplied (food) into energy released (work). Whether the second law of thermodynamics is equally applicable to human as to non-human processes is not as yet firmly established,[4] but the limitations of this method from the engineering viewpoint are:

a Difficulties in ascertaining the calorific value of the food consumed by workers

b Differences in individual coefficients of digestibility, assimilation, etc.

c Differences in individual energy consumption for physiological processes at rest (breathing, blood circulation, etc.)

d The time lag between energy consumption and energy release

e The impossibility of differentiating between energy release and fatigue for various jobs within the day's work

f The impossibility of differentiating between useful and useless expenditures of energy.

The value of these energetics researches is apparent from the results of experiments demonstrating that the work of one group of muscles is more efficient (thermodynamically) than that of certain other groups.[1] On the other hand, as J. B. S. Haldane has shown, a dose of about 7 grams of acid sodium phosphate increases a man's capacity for prolonged muscular work by about 20 per cent, and probably aids in prolonged mental work. Again, the psychological effects of fascination in work or worries of any kind markedly increase or reduce the capacity for work.

The chemistry of fatigue since the work of Du Bois-Reymond has received considerable attention.[2] Protein catabolism and other changes in the blood and urine have been studied. However, the increase of acid in urine and the higher concentration of the hydrogen ion of the urine are not a reliable measure of fatigue without an accurate knowledge of a man's diet. (A. Baird Hastings, 1919.[3]) Similarly, phenol production and excretion depend mory upon the habits and diet of men than upon their muscular activite (E. L. Scott and A. B. Hastings, 1920). Furthermore, most investigators agree that muscular work is not accompanied by an increased nitrogen excretion, although there may be a marked increase some hours later. Scott and Hastings[4] think that "this is in agreement with the lag of 24 hours reported by Atwater, Woods, and Benedict." Thus, for the purpose of studying engineering and administrative means of readjustment of the elements of daily work with the view of reducing fatigue, the composition of urine cannot be as yet conveniently used as an index of fatigue in industrial workers.

The attention of investigators was then turned from the study of the presence of certain substances in the organism produced by exertion to the mechanism of their removal. Thus, quoting A. B. Hastings,[5] "Loewy found that if the tissues were supplied with sufficient oxygen during muscular work, the respiratory quotient remained unchanged. Higley and Bowen, in a series of carefully controlled experiments, found that the rate of production of carbon dioxide remained constant for constant work." Inasmuch as the function of the organs of respiration is to renew the supply of oxygen in the depleted venous blood and to remove its increased carbon dioxide content, the respiratory quotient and the composition of exhalation vary with the necessity for pulmonary ventilation. This ventilation itself is "the result of excitation of the respiratory center by an increased hydrogen-ion concentration of the blood, by decreased blood flow, and by afferent impulses."[6]

These facts present for study two highly important questions:

1 Is the relation between the work done and the carbon dioxide exhaled such as to offer an index of fatigue?

2 If so, what is the technique of observations which will offer a simple and practical guide for management engineers in solving their problems?

Thus the laboratory and field researches of physiologists are presenting a practical problem to management engineers.

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[1] President, Walter N. Polakov & Co., Inc. Mem. A.S.M.E.

[2] Shop Management, F. W. Taylor, Trans. A.S.M.E., vol. 24. p. 1337.

[3] Mastering Power Production, W. N. Polakov. Eng. Mag. Pub. Co., N. Y., 1921, p. 254.

[4] Ueber die Anwendbarkeit des zweiten Hauptsatzes der Thermodynamik auf Vorgange im tierischen Organismus. Pfluger's Archiv., Band 171.

Contributed by the Management Division for presentation at the Annual Meeting, New York, November 30 to December 4, 1925, of THE AMERICAN SOCIETY OF MECHANICAL ENGINEERS. All papers are subject to revision.

[1] The work of the legs was found to be more economical than the work of the arms, etc. Langworthy and Barett found, for instance, that hemming on a foot-operated sewing machine requires per hour six times more energy than hand hemming but the expenditure of energy per yard is only one-half as great; while operating an electrically driven sewing machine requires one-third of the energy required by a foot-operated machine per hour and only one-fifth as much energy per yard.

[2] E. Du Bois-Reymond (Monatsber. d. Kgl. Preus. Akad. d. Wissensch., 1859, p.288) reported experiments which showed that a tetanized frog's muscle was more acid than an untetanized one.

[3] Also Sawicki: "The quantity of acid secreted by the urine depends more on the quality and quantity of the food ingested than on the rest and work."

[4] Some Phases in Protein Catabolism and Fatigue. Public Health Report No. 617, U. S. Treasury Department.

[5] An Investigation of Changes in Blood, etc. Public Health Report No. 546. U. S. Treasury Department.

[6] A. B. Hastings, The Physiology of Fatigue. Public Health Bul. No. 117.

Be sure to bring this copy with you to the Annual Meeting

The relation between the muscular work done by a man and the relative content of CO_2 in his exhalation, according to Waller's[1] early experiments, is fairly consistent as appears from Table 1. For more accurate observations it is necessary to take into consideration the weight of the worker and the surface of his body. The relation between these has been observed by Lee, Rubner, Lissauer, and others, and the average may be accepted as

$$S = 11.1\sqrt[3]{W^2}$$

where S is the surface of the body in square meters and W the weight in kilograms.[2]

When individual differences are constant, i.e., in observations on the same individual walking and running at different speeds, the net CO_2 exhalation and caloric discharge increase with the speed thus:[3]

		Net CO_2 per sec.	Kg-cal. per hour
Walking	3.4 miles per hour	11.0	232
Walking	3.9 miles per hour	17.2	362
Running	5.9 miles per hour	29.7	626
Running	7.2 miles per hour	51.8	1091

Experiments of Cathcart, Trafford,[4] Briggs,[5] and Waller,[6] while carefully conducted and valuable from the physiological point of view, are unfortunately lacking in details of work done, implements used, environment, and other most important engineering and managerial data; likewise their value is depreciated by their failure to note the progressive output of workers observed and to measure the accumulated fatigue. In the bottom section of Fig. 1 the results of Waller's observations for a week (beginning Saturday) on a dock laborer, Tom King, are reproduced graphically; unfortunately the nature of work is not described in necessary detail, and the output is not noted.

TABLE 1 CORRESPONDENCE BETWEEN WORK DONE AND CO_2 EXHALED[1]

Kg-m. per sec.	Thermal efficiency, per cent[2]	Kg-m. per cu. cm. of CO_2	CO_2, cu. cm. net
43.5	45.0	1.13	38.5
30.0	31.6	0.79	38.0
27.3	39.6	0.99	27.6
27.1	32.8	0.82	33.2
21.6	41.6	1.04	20.8
20.3	36.4	0.91	22.5
19.0	31.6	0.79	24.0
18.2	32.4	0.81	22.5
17.5	25.0	0.62	28.0
14.9	26.0	0.65	22.9
13.7	29.2	0.73	18.6
13.6	25.0	0.63	21.5

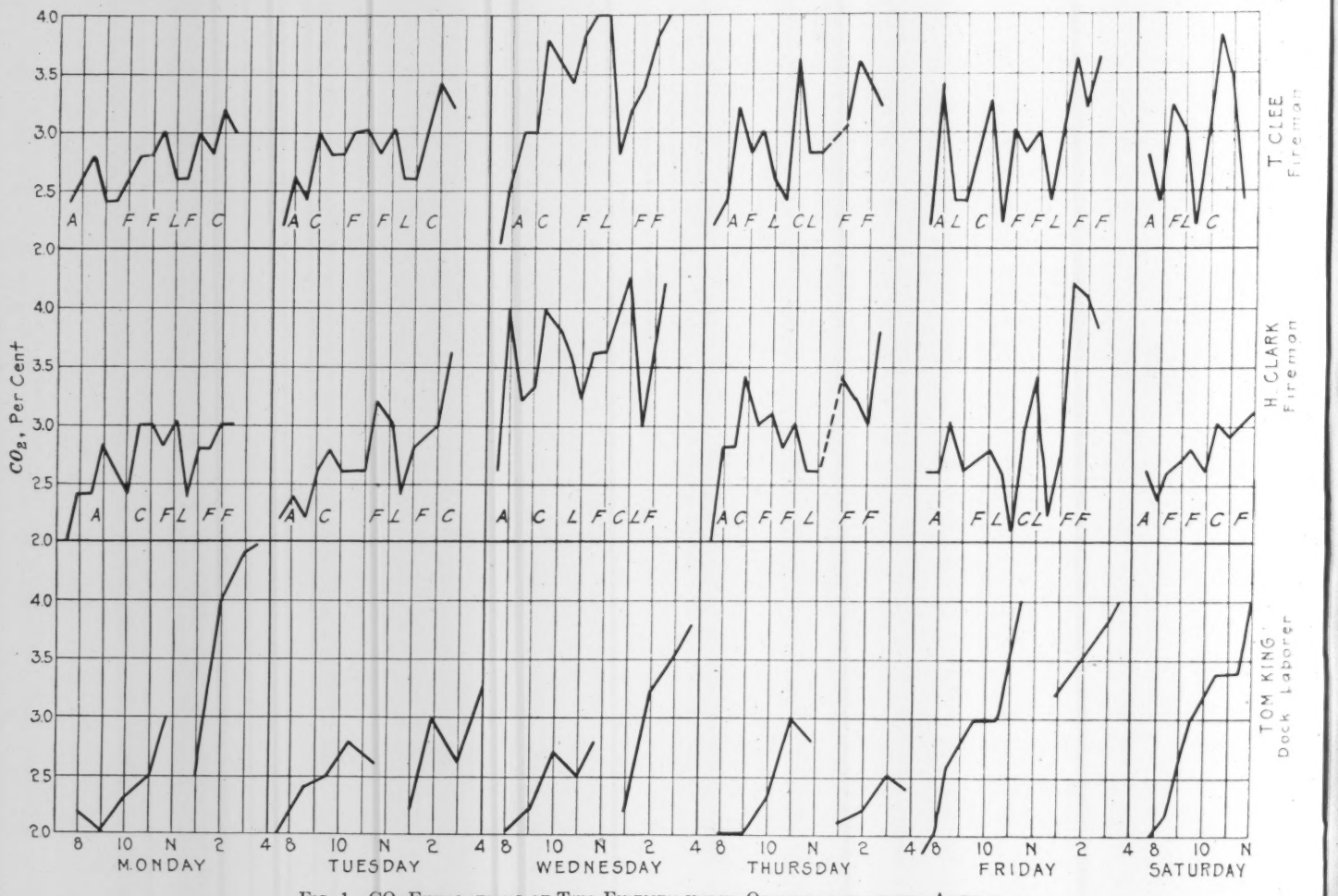

FIG. 1 CO_2 EXHALATIONS OF TWO FIREMEN UNDER OBSERVATION BY THE AUTHOR

A—Ash removed. C—Clinkers taken out. F—Fires worked or leveled. L—Lounging or lunch.

In the author's observations on two firemen, Clee and Clark (two upper sections, Fig. 1), both men were stoking two hand-fired 250-hp. boilers, firing per hour from 1700 to 2000 lb. of bituminous coal and using shovels holding 23 lb. The boilers carried a fairly uniform load of 125 per cent of rated capacity and obtained during the period of observation an equivalent evaporation of 10.46 lb.

[1] A. D. Waller, Work Measured by CO_2. *The Journal of Physiology.* Proc. Physiol. Soc., vol. 52, 1918–19–20.

[2] John R. Murlin, On the Law of Surface Area in Energy Metabolism: *Science*, Sept., 1921.

[3] A. D. Waller and S. De Decker, Physiological Cost of Walking. *Journal of Physiology*, vol. 56, 1921.

[4] E. P. Cathcart & F. I. Trafford, Energy Expenditure in Minor Duties. *Journal of Physiology*, vol. 53, no. 6.

[5] Henry Briggs, Physical Work and the Human Machine. *Engineering and Industrial Management*, 1921, vol. 6, no. 12.

[6] A. D. Waller, Calibration of a Dock Labourer by Means of his CO_2 Discharge. *Journal of Physiology*, vol. 52, no. 5.

[2] The second column shows the ratio of the mechanical work done to the energy consumed. Inasmuch as both are expressed in calories, it indicates the thermal efficiency of the labor process, the maximum being 45 per cent with 55 per cent converted into heat, while the minimum is 25 per cent of energy transformed into work and 75 per cent into heat and other bodily processes.

of water from and at 212 deg. fahr. with coal containing 14,100 B.t.u., i.e., maintaining the thermal efficiency of the boiler plant at 72.0 per cent. Ashes were removed from the ashpits by means of long-handled shovels once a shift (work hours from 7 a.m. till 3 p.m.), while the fires were worked and leveled and clinkers removed during the shift as conditions demanded. These operations were noted and marked on Fig. 1 with symbols as follows:

A Ash removed
C Clinkers taken out
F Fires worked or leveled
L Lounging or lunch.

The samples of exhaled gas were taken every 30 minutes by direct exhalation into the hand air pump and by collecting the pure sample of 50 cc. in the hand CO_2 analyzer. The arrangement was as shown in Fig. 2, the men gladly coöperating. In order to obtain data on the relation existing between the nature of work done and the gas metabolism, the author had to sacrifice volumetric measurements as too cumbersome for actual shop practice. This, however, was not deemed detrimental as he intended merely to develop a handy shop method which would serve with a sufficient approximation two purposes:

1 To establish a simple, practical method applicable in work-a-day factory environments
2 To use this method for the strictly utilitarian purpose of so modifying existing working habits as to reduce the strenuous elements of productive work.

With such an over-simplified method he was able to study work as it is actually performed in shops and factories, without any interference with the routine of work, at frequent intervals, over a large number of workers, and free from the influence of the artificial conditions of laboratory researches.

In order to further investigate the accumulation of fatigue of industrial workers over a longer period of time (week as a unit), the author combined a number of such observations made on workers in various industries at different occupations and secured an average closely corresponding to his observations on firemen. These data were plotted against the curve of power consumption from a public-utility plant in central Pennsylvania,[1] based on the average annual records of Monday's load by hours, Tuesday's load by hours, etc., as used by a large variety of industrial plants which were customers of this central station.

From this very instructive comparison of curves showing exhaled CO_2 (broken-line) and electric power used (solid-line), it appears that (see Fig. 3):

a On Mondays the worker is still under the influence of the incitement and is slow to develop the maximum of output in the morning hours but "warms up" in the afternoon. CO_2 (uncorrected for body functions) reaches 3 per cent and is fairly constant.

b On Tuesday the productive work reaches its maximum, while the rested workman exhales a minimum of CO_2, with an average of 2.7 per cent.

c On Wednesday the accumulation of fatigue begins to show, as work is slow to reach its peak in the morning and declines in the afternoon; the average CO_2 rises to 3.5 per cent, with peaks reaching and exceeding 4.0 per cent.

[1] Walter N. Polakov, Mastering Power Production. Fig. 51. Cecil Palmer. London.

d Thursdays indicate further effects of fatigue in the industrial community characterized by only one peak of short duration in the morning and low, sluggish work in the afternoon, while the CO_2 exhaled is also on a decline, showing that the workman is "sparing himself." The correspondence of ups and downs of both output and CO_2 curves is suggestive.

e Fridays are most typical of exhaustion of the vitality of the industrial community. After a determined effort in the morning of very short duration, the work falls off from hour to hour steadily, from whistle to whistle, with even the lunch recess bringing no rest or new vitality. At the same time the CO_2 curve is most interesting as it shows retarded metabolism in the morning only slightly above what would be expected from men at rest, while in the afternoon when efforts to spur up the work are made, the percentage of CO_2 reaches sharply 4.2 per cent and then only gradually declines, leaving the impression of continual exertion of fatigued men.

FIG. 2 METHOD OF COLLECTING EXHALED CO_2

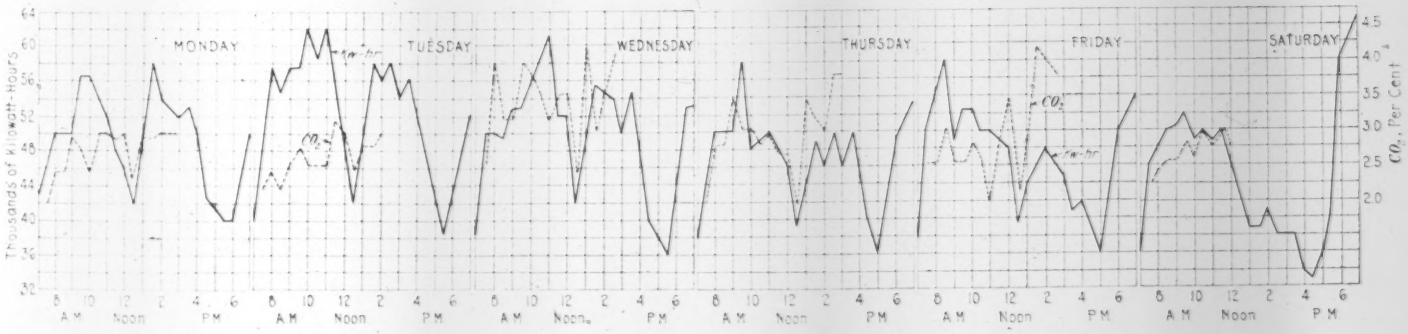

FIG. 3 AVERAGE POWER LOADS ON A CENTRAL STATION AND AVERAGE CO_2 CONTENT OF FIREMEN'S EXHALATIONS

f Saturdays again show the surprisingly close relation between spurts in work and high peaks in CO_2 exhaled.

These findings are in the main similar to the observations of Waller and De Decker[1] in the printing plant of the London *Times* on three different occupations. See Table 2. If these are compared with the author's data obtained with firemen, it becomes

TABLE 2 CUBIC CENTIMETERS OF CO_2 EXHALED PER SECOND

Hour	Compositor	Type foundry	Page setters
5 P.M.	2.5	3.1	...
6 P.M.	4.8	..	...
7 P.M.	5.9	5.0	...
8 P.M.	6.1	6.5	2.6
9 P.M.	7.8	7.9	...
10 P.M.	9.8	10.0	6.19
11 P.M.	11.7	13.3	9.58
12 midnight		15.4	13.00
1 A.M.	12.8	18.2	13.40
2 A.M.	..	..	18.9
3 A.M.	..	..	20.4
4 A.M.	..	..	...

[1] A. D. Waller and I. De Decker, The Physiological Cost of Work in Various Departments of *The Times* Printing House. *Journal of Physiology*, vol. 53, no. 6, 1920.

evident that the repetitive and uniform operations of compositors and page setters is reflected in a steady rise of the CO_2 exhaled, while sporadic exertions of firemen at physically strenuous operations intermingled with less arduous coal shoveling and rest periods manifest themselves in fluctuating percentages of the CO_2 in exhalation.

Inasmuch as the acquisition of skill at mechanical operations corresponds with more precise motions and the elimination of useless strain and uncoördinated movements, it was expected that the CO_2 in exhalation might also serve as an index of the successful training of workers at manual operations. In training firemen the author employed a "dummy" furnace consisting of a platform 10 ft. deep, 12 ft. wide and 26 in. above the floor level, with frames in front representing the firing doors. With this device he has trained unskilled men in the art of throwing coal into the furnace, first evenly spreading it, and then covering precisely the required spots. After some practice the men were blindfolded to demonstrate that the accuracy of the motions did not depend on the aid of the sight, as this would be a little help in a blazing furnace. The results obtained with a green man from the farm after one hour of training and with the same man six weeks later were as follows:

CO_2 in exhalation after first hour of training.....4.3 per cent
CO_2 in exhalation after one hour after 6 weeks.....2.7 per cent

Waller and De Decker, studying the effect of training in walking,[1] reported even more pronounced conservation of energy due to training:

Before training in walking.....0.215 cu.cm. net CO_2 per kg-m.
After two months of training.....0.098 cu.cm. net CO_2 per kg-m.

From these studies the author feels warranted in drawing the following preliminary conclusions:

1 Maximum CO_2 in exhalation is an index of full load on the muscles
2 Decrease of CO_2 content in spite of continued work indicates overload and is a signal of fatigue
3 Age and size of the worker influence the CO_2 content. The younger and lighter men produce more work with the same CO_2 discharge than the older and heavier men
4 Peaks in the CO_2 curve immediately follow exertions
5 Training of workers considerably reduces CO_2 exhalation for the same release of productive work
6 The efficiency (biological) of work decreases and the expenditure of energy per unit of work increases as work continues (per day or week).

The conclusion cannot be escaped that the convenience, simplicity, and accuracy of indications obtained by the CO_2 analysis method offer a means heretofore unavailable to reduce the fatigue resulting from (necessary) work motions by readjusting operations or by providing labor-saving means to reduce CO_2 peaks. It further offers an unmistakable means of judging whether men are properly trained to conserve their energy while working. It likewise offers a more reliable guide for placing men and women in jobs which will not overtax them physically. Thus management, foremen, instructors, and personnel supervisors will gain much by using this method and by substituting facts for mere judgment. They will save human energy, thus enhancing the health of the workers and the productivity of labor.

[1] See footnote 14.

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The Development of a Unit Pulverizer

Description of a Machine for Preparing Coal to Be Burned in Powdered Form and of Its Development—Results of Tests to Determine Its Reliability, Ability to Pulverize Wet Coal, Power Consumption, Durability, Fineness of Grinding, Etc.

By R. SANFORD RILEY[1] and OLLISON CRAIG,[2] WORCESTER, MASS.

THIS paper covers the development of a unit pulverizer for preparing coal to be burned in powdered form. It does not touch on furnace design or furnace performance, but only no the action of the pulverizing elements. Even the feed mechanims of the pulverizer is not discussed in detail because it is not a pulverizing problem. The term "unit pulverizer" is used to define that type of pulverizer which discharges the pulverized product directly into the furnace where it is burned; as distinguished from a pulverizer which discharges into a "bin" or "storage system."

Development began with a machine manufactured in England by Alfred Herbert Ltd., known as the "Atritor," size No. 16, designed to operate at 1200 r.p.m. with a nominal capacity of 4000 lb. of coal per hour. The machine was the invention of Charles E. Blyth, a recognized expert on fine pulverizing for cement manufacture.

Certain characteristics are desirable in a unit pulverizer and it was believed that these characteristics could be obtained in the Atritor because of the principles incorporated in it. For this reason it was selected for study and development.

Description of the Original Machine and Its Action

The construction of the machine as received from England is shown in Fig. 1, with parts cut away to show the interior. A main shaft projects horizontally through a housing, the housing being divided into two chambers. The shaft carries a disk which revolves within the first, or pulverizing chamber. The second chamber contains a fan rotor carried on the same shaft as the disk. The revolving disk contains concentric rows of pegs projecting from each side. These revolving pegs alternate with similar rows of stationary pegs which are attached to the disks forming the sides of the pulverizing chamber. The shell or periphery of the chamber fits around the disk so as to form an annular passage from one side of the disk to the other. The pulverizing chamber and the fan chamber are connected by a passage concentric with the shaft. Blades, called "rejector" blades, are mounted on the shaft and revolve within the entrance to this passage.

The fan draws air through the pulverizer in sufficient quantity to act as a carrier for the coal to be pulverized. When grinding at the rate of 4000 lb. of coal per hour, this quantity of air represents about 10 per cent of the total air required for combustion. Coal is fed into the machine near the shaft on the side of the disk opposite the fan, or the side which is called the "first effect." The revolving pegs and the flowing air carry the coal past the successive rows of moving and stationary pegs to the edge of the disk. The air flow then carries the coal through the annular passage formed by the disk and the housing to the side of the disk adjacent to the fan. The space on this side is called the "second effect." After making the turn around the edge of the disk, the air carries the coal past the successive moving and stationary pegs in the second effect, past the rejector blades, through the passage between the pulverizing and fan chambers, through the fan, and out the discharge.

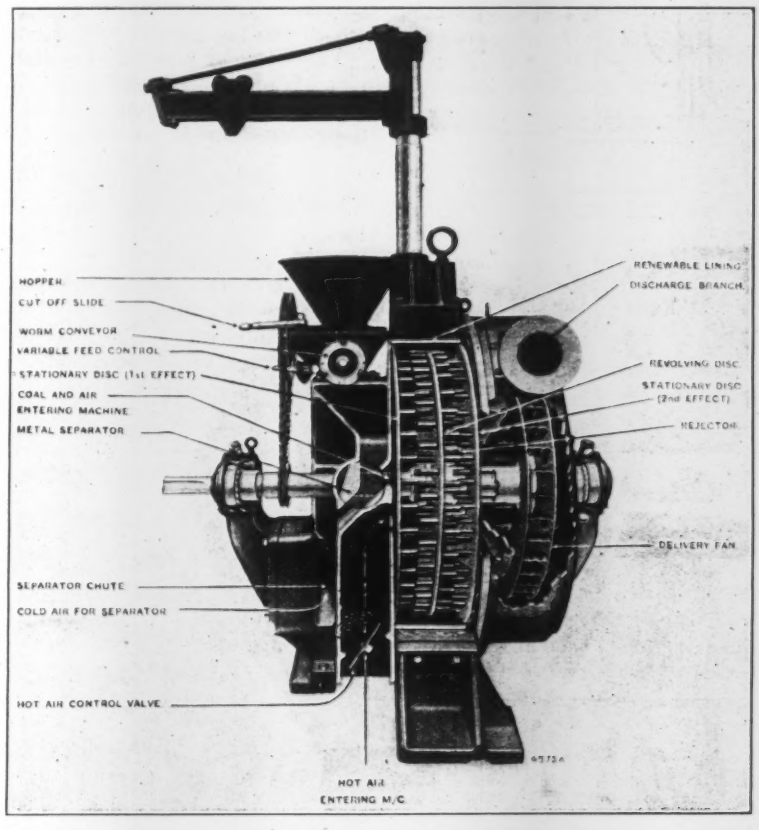

FIG. 1 INTERIOR OF ATRITOR, SHOWING METAL SEPARATOR, FIRST AND SECOND PULVERIZING EFFECTS, HOT- AND COLD-AIR PASSAGES, AND FAN

In the first effect the coal is crushed by impact of the moving pegs and also by impact against the stationary pegs. In addition to this there is a violent turbulence of the air between the moving pegs which further serves to pulverize the coal by a process of attrition on itself. In passing through the first effect the coal is only coarsely pulverized, and no separation occurs.

Fine pulverization is accomplished in the second effect, where the action produces a separation of coarse and fines as well as pulverization. In this second effect the moving pegs produce a fan action which tends to blow outward contrary to the flow produced by the fan. Violent eddy currents are set up around the moving and stationary pegs by these counteracting forces, which rub coal particles against other coal particles. The centrifugal force of the revolving mass of coal retards the passage of the large particles of coal and produces the separating effect. Since the propelling forces produced by the fan are greater than those produced by the moving pegs, the resultant motion is from the periphery of the disk to the center and out of the pulverizing space into the fan inlet, regardless of what intermediate motions occur.

In Fig. 2 an attempt is made to indicate the action of the air currents between stationary and moving pegs. Revolving pegs are marked *A* and *C*, stationary pegs, *B* and *D*. The arrows in this cross-section are drawn thus to suggest eddy currents or turbulence. It will be noted that there is a substantial clearance between pegs, not less than $\frac{1}{2}$ in., so that close approach or contact is not depended on for pulverization.

The stream of air and material induced by the fan is shown by the arrows in Fig. 2, following closely around the interior surface of the pulverizing chamber. The stream of air and material developed by the pegs is shown by arrows near the revolving disk. Between these two streams are looped arrows indicating eddies

[1] President, Riley Stoker Corp. Mem. A.S.M.E.

[2] Engineering Department, Riley Stoker Corp. Mem. A.S.M.E.

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which no doubt have a great influence on the attrition of the material. Attention is directed to the action in the second effect because this seems to be novel and less understood.

There have been many pulverizers constructed with pegs on the side of a moving disk alternating with fixed pegs, but the flow of the material is usually outward as in the first effect just described. It is the inward flow against centrifugal force that calls for special examination of the action that gives both pulverization and separation. Great significance is attached to the relative length of the pegs or "beaters" in the second effect. It will be noted that the moving pegs are relatively short in comparison with the stationary pegs. The stationary pegs reach approximately across the entire width of the second effect, whereas the revolving pegs reach across approximately one-half this distance.

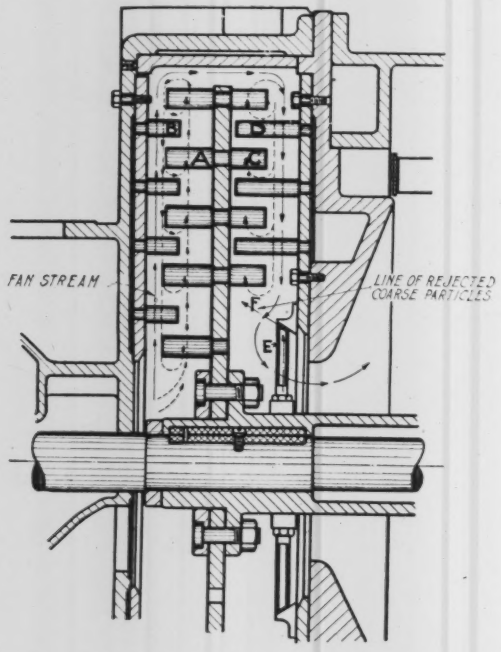


FIG. 2 ACTION OF AIR CURRENTS BETWEEN STATIONARY AND MOVING PEGS

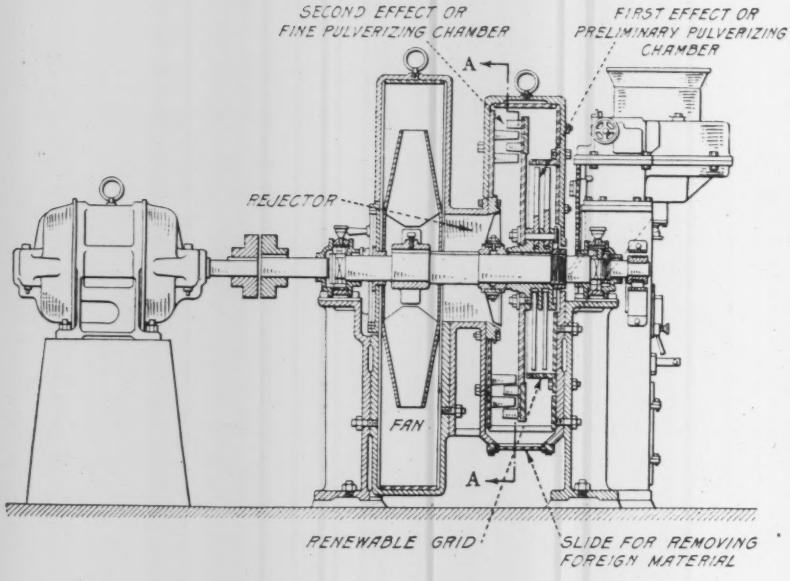


FIG. 3 SHOWING FINAL DESIGN OF PULVERIZER IN SECTION

The fan stream as indicated by the arrows follows closely the stationary side wall of the chamber, and passes along the shaft toward the fan discharge chamber. The currents of air and material set up by the revolving pegs are outward from the center, hence we have the two opposing streams in different parts of the pulverizing chamber. Obviously the air and material adjacent to the revolving disk are under the influence of centrifugal force, while both adjacent to the stationary disk are under the influence of the fan which draws everything toward the central discharge opening. Between these two influences there are eddy currents with violent turbulence.

Of course it is understood that in addition to the axial currents indicated on the drawing, there are peripheral currents due to the revolution of the disk and pegs. The resultant currents set up are really made up of three components, namely, centrifugal force, fan suction, and revolution of the revolving pegs. Obviously, the particles most affected by the fan stream are those of smallest size, hence the fine products are being continuously drawn off by the fan action. The larger particles are more affected by the blows from the revolving pegs, and herein lies the combination of pulverization and separation.

As the products of pulverization reach the shaft, they are subject to a further separating action by the revolving arms or rejector blades *E*. Heavier particles are thrown outward against the edge of the discharge orifice and then deflected as shown by the arrows *F* toward the revolving disk. As they approach the revolving disk they come under its influence and are thrown outward from the center into the eddies above described.

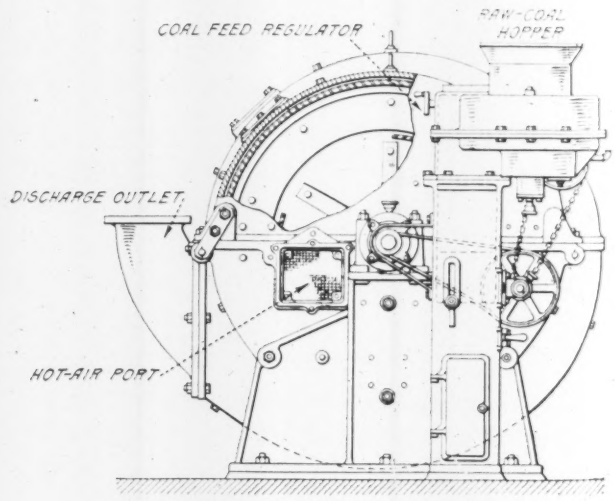
The foregoing is simply an attempt to visualize what happens in view of the construction shown. Different theories of action might be evolved, but it is believed that the above statement is substantially correct and in accordance with results obtained.

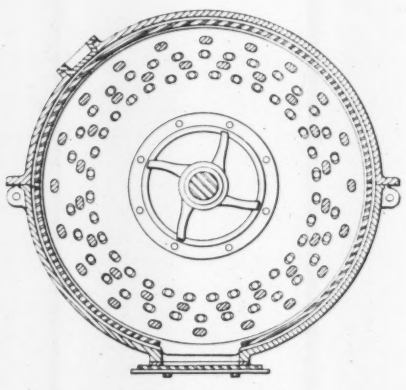
CHARACTERISTICS OF THE ORIGINAL MACHINE

The machine was examined and studied for the following characteristics:

1 Reliability
2 Ability to pulverize wet coal
3 Power consumption
4 Maintenance
5 Fineness of pulverization.

This order was considered also the order of importance or desirabil-





SECTION A-A

ity in a pulverizer. Trials of the machine during a period of one year showed the following results.

1 Foreign material occasionally entered the machine, causing breakage of pegs. The machine was equipped with a separator for the purpose of removing foreign material, but the separator was not infallible with American coals.

2 Ability to pulverize was only slightly affected by quantity of moisture in coal as the coal was pulverized in suspension in an air current which prevented packing or caking. Means were also provided for using heated air, which acted as a drier to the coal.

3 Preliminary power tests indicated a power consumption about equal to, or slightly greater than, other unit pulverizers.

4 The pegs were made of gray cast iron. This material did not permit sufficient life before renewal for economical maintenance. The machine was split vertically, and considerable time was required to change the disks containing the pegs, and also to replace the pegs.

5 The machine pulverized coal to an extreme fineness, tests of a large number of samples showing from 90 per cent to 97 per cent through a 200-mesh screen with practically 0.0 per cent remaining on a 50-mesh screen.

CHANGES INDICATED

A study of the results obtained with the machine indicated that:

a The machine should be so designed as to be proof against breakage by entrance of foreign material

b Unnecessary fineness should be reduced so as to decrease power consumption

c The life of the pegs should be increased and renewals made easier.

CHANGES MADE AND DESCRIPTION OF FINAL MACHINE

Fig. 3 shows the final machine in section. Two chambers are

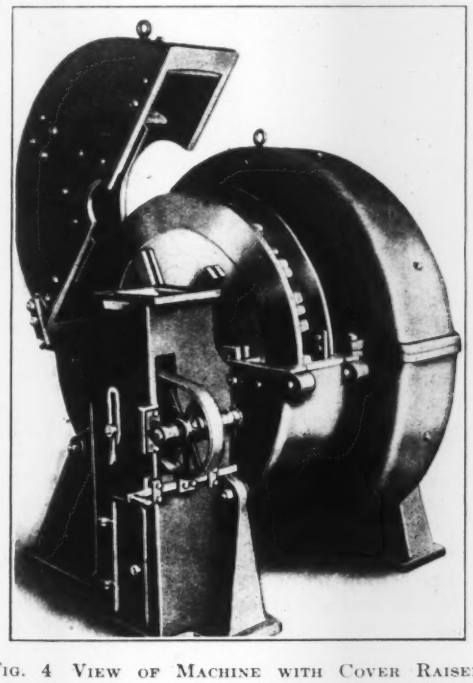


FIG. 4 VIEW OF MACHINE WITH COVER RAISED

provided as before, one a pulverizing chamber and the other a fan chamber, the two being connected by a passage concentric with the shaft. The shaft still carries a disk, a fan, and rejector blades in the passage between the two chambers. All pegs are removed from the first effect and replaced by swing hammers revolving within a grid. The grid circle, however, is not complete, one section being omitted. The number of moving pegs in the second effect was reduced from four rows to two rows, and the stationary pegs in the second effect were reduced from three rows to two rows. The pulverizing chamber and fan chamber are split horizontally and so arranged that the pulverizer chamber can be opened without disturbing the cover to the fan chamber.

The pegs are made of a special hard-composition iron. Tests indicated that this material lasted longer in use than any of the well-known alloy steels generally used for resistance to abrasion. The base of the pegs is formed by a pressed-steel ferrule and the iron cast into the ferrule. The ferrule formed a flanged base. The moving pegs are held in position by clamping the base flanges between the steel disk and a steel ring, the flanges of the stationary pegs are clamped between the stationary plates forming liners for the pulverizing chamber, and the housing. Casting the pegs in a pressed-steel ferrule and utilizing the steel flange so formed as a means for fastening pegs, allows a material and a shape of peg to be used which could not be machined or ground, and removes any restrictions on peg design, material, or life which might otherwise be imposed by the necessity of machining or grinding.

Figs. 4 and 5 show the top of the pulverizer chamber after exposing to view the arrangement of hammers, disk, and pegs.

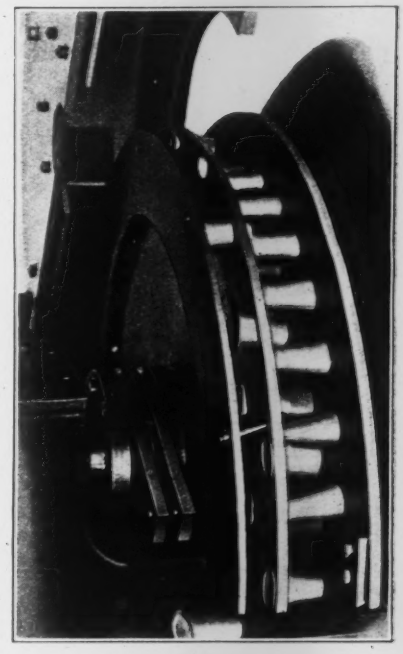


FIG. 5 VIEW OF PULVERIZING CHAMBERS WITH COVER LIFTED AND RETAINING RING LOOSENED TO SHOW HOW PEGS CAN BE REPLACED

DESCRIPTION OF POWER TESTS

In making an analysis of the power required to pulverize by a machine, it was assumed that the total power of pulverization was the sum of two separate parts, one part being a fixed or constant power for any given machine operating under any given condition, and the other part being a variable, constantly increasing with the grinding rate.

Referring to Fig. 6, it was assumed that a straight line M would represent the total power in kilowatts for any rate of pulverization, and that for any rate of pulverization the total power in kilowatts would be represented by the constant quantity a plus the quantity b. The quantity b is obtained by multiplying the rate of pulverization W by the rate of increase in power required per ton of coal s. If P represents total power at any rate, W, then

$$P = a + b = a + sW \quad [1]$$

The horizontal line N representing the constant power was assumed to intersect the total power line M at the point of zero rate of pulverization.

The power rate in kilowatt-hours per ton of coal pulverized represented by the curve R was obtained by dividing the total power for any point on M by the corresponding pulverization rate in tons per hour. If p represents the power rate in kilowatt-hours for the pulverization of one ton of coal at any rate of pulverization, then

$$p = \frac{P}{W} = \frac{a}{W} + \frac{sW}{W} = \frac{a}{W} + s \quad [2]$$

Equation [2] represents the curve R and indicates that the power rate per ton at any rate of pulverization is a first-power function of both a and s. Differentiating [2], we obtain

$$dp = \frac{Wda - adW}{W^2} + ds = \frac{adW}{W^2} \quad [3]$$

whence

$$\frac{dp}{dW} = \frac{a}{W^2} \quad [4]$$

or the flatness of the power-rate curve R is dependent on the magnitude of the constant power a.

The constant power in a pulverizer is probably made up of bearing friction, windage, and propulsion of air. The variable power is essentially the work expended in coal pulverization and propulsion.

A program for a series of tests was drawn up. Successive tests with various pulverizer combinations were planned and it was expected that each combination would produce a condition in which resistance to passage through the machine would be lessened, through reducing the factor a, and in which less work would be done on the coal, thus reducing the factor b but resulting in coal of less fineness.

The changes contemplated were as follows:

1 Replacing two rows of moving and one row of stationary pegs in first effect with six swing hammers surrounded by a grid circle. The grid circle consisted of twenty cast segments, each segment occupying an arc of a circle about five inches in length. The grids were

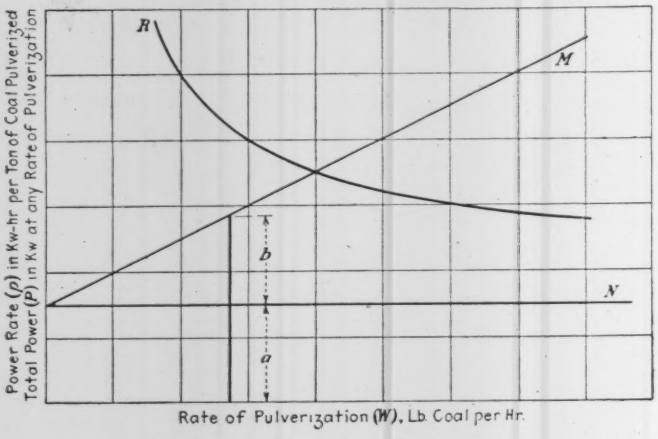

R
M
N
b
a
Power Rate (p) in Kw-hr per Ton of Coal Pulverized
Total Power (P) in Kw at any Rate of Pulverization
Rate of Pulverization (W), Lb Coal per Hr.

FIG. 6 DIAGRAMMATIC REPRESENTATION OF RELATION OF RATE OF PULVERIZATION TO POWER RATE PER TON OF COAL PULVERIZED AND TO TOTAL POWER AT ANY RATE OF PULVERIZATION

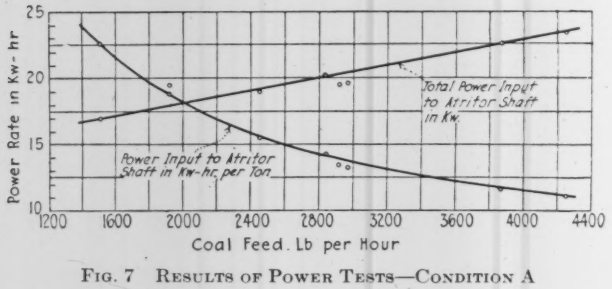


FIG. 7 RESULTS OF POWER TESTS—CONDITION A

made with spaces cast in for passage of coal which had been sufficiently reduced by the hammers.

2 Omission of a number of grid segments to provide large openings for the passage of coal.

3 Elimination of all pegs from first effect.

4 Elimination of a portion of the pegs from second effect.

5 Elimination of rejector blades.

6 Increase of clearance around disk periphery by removing liner from grinder housing.

A series of power tests were made for each of a number of combinations of above changes in machine condition. For each test a weighed quantity of coal, from 500 to 800 lb., was pulverized. This coal was placed in a small hopper above the machine feeder and the time required to pass this quantity of coal through the feeder taken by a stop watch. Start and stop conditions were taken at a point where the coal passed a circular section about 15 in. in diameter. A start-and-stop error of 1 in. at this section would result in an error of 4 lb. or approximately one-half of one per cent in the determination of rate of pulverization.

The pulverizer was driven by a 60-hp. slip-ring motor. Power readings were taken from a wattmeter at one-minute intervals during each test.

The coal-weighing larry and the wattmeter were carefully calibrated both before and during the series of tests.

During each test a sample of raw coal was taken at the pulverizer feeder, and a sample of pulverized coal was taken by an aspirator from the nozzle of the burner. The burner was of the mixing type and the pulverized sample was taken after the point of mixing to insure an average sample. All samples were immediately placed in glass jars with rubber sealing rings. Both raw-coal and pulverized-coal samples were tested for moisture, and the pulverized samples tested for fineness in a Ro-Tap screen testing machine. Each series of tests was designated by a letter and each series of tests was made for a particular combination within the machine. The combinations and the designating letters are tabulated below.

Symbol representing arrangement	Arrangement in Machine
A	Swing hammers, all grids, 3 rows moving pegs and 2 rows of stationary pegs in first effect. All pegs in second effect. Rejector blades out.
B	Swing hammers and all grids in first effect. No pegs in first effect. All pegs in second effect. Rejector blades out.
C	Swing hammers in first effect. No pegs in first effect. Five alternate grids removed from top half of grid circle. All pegs in second effect. Four rejector blades on shaft.
D	Swing hammers in first effect. No pegs in first effect. Five alternate grids removed from top half of grid circle. All pegs in second effect. Two rejector blades on shaft.
E	Swing hammers in first effect. No pegs in first effect. Alternate grids removed from entire grid circle. Two rejector blades on shaft.
F	Swing hammers in first effect. No pegs in first effect. Alternate grids removed from entire grid circle. Four rejector blades on shaft.
G	Original Atritor condition. All pegs in both first and second effect. No swing hammers used. Four rejector blades on shaft.
H	Same condition as C.
J	Swing hammers in first effect. Five alternate grids removed from top half of grid circle. No pegs in first effect. Two rows moving and two rows stationary pegs in second effect. Four rejector blades on shaft.
K	Swing hammers in first effect. Five alternate grids removed from top half of grid circle. No pegs in first effect. Two rows moving and two rows stationary pegs in second effect. No rejector blades on shaft.
L	Same condition as K.
M	Swing hammers in first effect. Five alternate grids removed from top half of grid circle. No pegs in first effect. Two rows moving and two rows stationary pegs in second effect. Two rejector blades on shaft.
O	Swing hammers in first effect. Five alternate grids removed from top half of grid circle. No pegs in first effect. Two rows moving and two rows stationary pegs in second effect. No rejector blades on shaft. Liner removed from pulverizer housing to increase area of passage at periphery of disk.

For each series of tests a curve sheet was constructed, using rate of pulverizing in pounds of coal per hour for abscissas, and kilowatts for total-power curves and kilowatt-hours for power-rate curves as ordinates. Fig. 7 gives the curves for arrangement A described above.

The power quantities in kilowatts and power rates in kilowatt-hours as indicated on the ordinates of the curves are in each case power at the Atritor shaft and not power input to the driving motor. When the original machine was first installed, a 60-hp. motor was considered necessary and this motor was used during all the tests. However, when lower power consumption was realized, the motor was much larger than necessary and operated inefficiently. To correct for this, calibration curves for the motor were made by using an Alden dynamometer and measuring input to and output from the motor. From the data thus obtained a power input-output curve and motor-efficiency curve were drawn and these curves used for determining the output of the motor, or input to pulverizer, for any given input to the motor. These curves are shown in Fig. 8. It will be noted that the motor efficiency during some tests was only about 50 per cent.

In constructing the curves, the total power curve was drawn first in each case by plotting the points from the data obtained. In every case a straight line satisfied the points as closely as any curve that could be drawn. The equation of these lines is given by [1]. For each rate of pulverizing the power per ton was calculated by dividing the total power by the grinding rate in tons and the calculated points plotted and curves drawn through these points. Points obtained by calculation from test data were then plotted, and in each case the curves drawn satisfied the points obtained by

tests. These curves are represented by Equation [2], and Equation [4] indicates the flatness of these curves.

DISCUSSION OF TESTS AND RESULTS

In Fig. 9 all the power-rate-per-ton curves are plotted for comparison, and to visualize the reduction in power by successive changes in the Atritor.

Table 1 gives the essential data for all series of tests.

Fig. 10 is a curve drawn with the constant power a for each test series as abscissas and flatness factors as ordinates. The flatness factor was taken for each series as the difference in power per ton at a grinding rate of 4000 lb. per hour and at a grinding rate of 1500 lb. per hour. The curve may be represented by an expression $\frac{\Delta p}{\Delta W} = f(a)$, and, as would be expected, this curve is a straight line passing through zero.

The data obtained indicated in general that omission of alternate grid sections in the upper half of the grid circle resulted in lower power consumption than with all grids in place or with alternate grid sections omitted from the entire circle. With all grids in position there was greater resistance to passage of coal and air than when larger openings were provided by omission of sections. If sections were omitted from the lower portion of the circle, most of the coal fell into the lower portion of the pulverizing chamber and only a portion of the disk periphery was effective in pulverizing. With omission of sections from the upper portion of the circle only, the coal was thrown upward in the pulverizing chamber and carried uniformly around the edge of the disk by the air stream, thus making the whole disk periphery effective.

Omission of pegs from the first effect, reduction of pegs in the second effect, and removal of rejector blades all resulted in lessened resistance through the pulverizer, less work done on the coal, and reduced fineness.

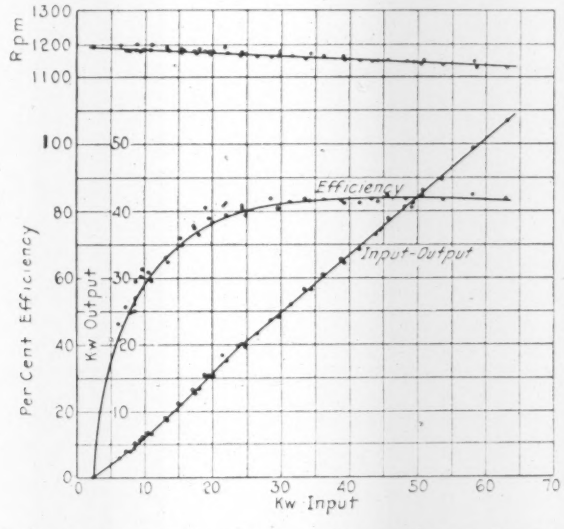


FIG. 8 CURVES FROM 60-HP. INDUCTION MOTOR USED IN TESTS OF PULVERIZER

CONCLUSIONS

A survey of the data obtained indicated that condition J gave the best compromise between power consumption, flatness of power curve, degree of fineness, and uniformity of fineness. The present machine was designed to meet condition J and is shown in Figs.

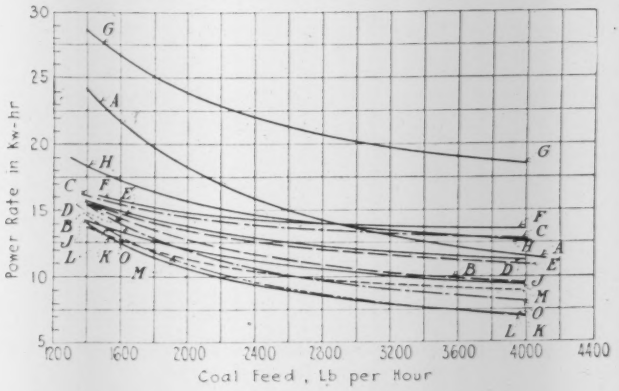


FIG. 9 POWER-RATE CURVES OF TESTS A TO O, INCLUSIVE
(Tests made March 31 to May 27, 1925.)

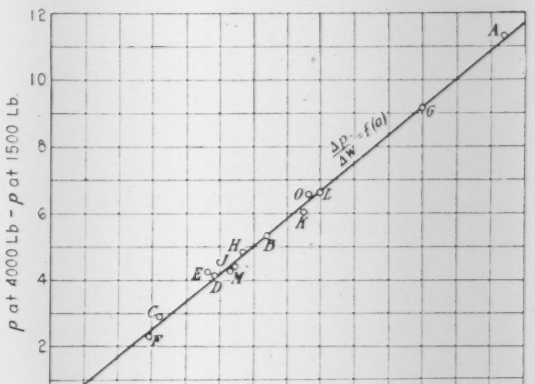


FIG. 10 CURVE SHOWING RELATION BETWEEN CONSTANT POWER QUANTITY (a) AND FLATNESS OF POWER-RATE CURVES

3, 4, and 5. The grid around the swing hammers was made of steel sections in place of cast sections. There is a single opening in the grid circle on the upper side in place of five openings as in condition J. The same upward spraying effect is produced by this arrangement and the opening is of sufficient size to allow any foreign material which can pass the feeder to pass out of the grid circle.

The steel grids have small circular holes spaced at intervals to permit passage of small coal particles at other points than through the large opening. The size of these holes was not determined by any degree of fineness to be obtained but they were merely made sufficiently large to prevent clogging by moist coal. The large opening in the grid circle is almost diametrically opposite the point

TABLE 1 DATA OF TESTS ON A UNIT COAL PULVERIZER[1]

Test	Moisture in raw coal, per cent	Moisture in pulverized coal, per cent	Moisture evaporated as per cent of coal	Temperature of air at hot-air inlet, deg. fahr.	Temperature of air at pulverizer discharge, deg. fahr.	Total power in kw. to Atritor shaft at 4000 lb. per hr.	Total power in kw. to Atritor shaft at 1500 lb. per hr.	Total power in kw. to Atritor shaft at Difference	Power rate in kw-hr. per ton to Atritor shaft at 4000 lb. per hr.	Power rate in kw-hr. per ton to Atritor shaft at 1500 lb. per hr.	Power rate in kw-hr. per ton to Atritor shaft at Difference	Constant power (a)	Variable power (b)	Screen fineness tests: Average per cent through 200-mesh	Screen fineness tests: Average per cent through 100-mesh	Screen fineness tests: Average per cent on 50-mesh	All grid sections in place	5 upper grid sections removed	5 upper and 5 lower grid sections removed	Pegs in first effect	No pegs in first effect	All pegs in second effect	Two rows of pegs in second effect	4 rejector blades in	2 rejector blades in	No rejector blades in	No liner in grinder housing
G	8.96	2.77	6.19	77	78	36.5	20.5	16.0	18.3	27.4	9.1	11.8	12.8	90.7	98.5	0.6											
F	3.23	1.00	2.28	77	82	26.8	11.7	15.7	13.4	15.7	2.3	2.9	12.0	93.1	98.6	0.3			—		—	—		—			
H	7.64	2.40	5.24	81	77	25.3	13.0	12.3	12.7	17.0	4.3	5.7	9.9	93.1	97.7	0.7											
C	7.43	0.83	6.60	332	...	25.3	11.6	13.7	12.7	15.6	2.9	3.3	11.0	97.4	99.1	0.3		—	—		—	—		—			
A	6.40	0.54	5.86	231	...	22.8	17.0	5.8	11.4	22.7	11.3	13.4	4.7	96.0	98.2	0.5	—			—					—		
D	7.74	1.18	6.56	85	103	22.1	11.4	10.7	10.8	15.1	4.3	4.9	8.6	92.1	98.8	0.3		—			—	—					
E	7.26	0.81	6.45	252	105	21.6	11.1	10.5	10.7	14.9	4.2	4.7	8.4	93.8	97.9	0.5			—		—	—					
B	9.00	1.48	7.52	75	...	19.0	12.4	6.6	9.5	14.8	5.3	6.4	6.3	93.0	98.0	0.4	—				—	—			—		
J	7.03	1.26	5.77	243	105	18.4	10.3	8.1	9.2	13.6	4.4	5.4	6.5	90.6	97.7	0.4		—	—		—	—					
M	6.06	1.71	4.35	245	106	17.4	9.8	7.6	8.7	13.0	4.3	5.3	6.1	86.0	96.4	1.1		—	—		—	—			—		
O	7.07	3.48	3.59	122	85	15.9	10.8	5.1	8.0	14.5	6.5	7.7	4.1	75.8	91.2	2.3		—	—		—	—			—	—	—
K	6.36	1.30	5.06	267	106	13.3	9.8	3.5	7.1	13.1	7.1	7.5	3.0	81.5	94.1	1.1		—	—		—	—			—		
L	6.87	1.75	5.12	130	81	13.8	10.2	3.6	7.0	13.6	6.6	8.0	3.0	74.5	92.8	1.4		—	—		—	—			—		

[1] Dashes in the columns at the right of the table confirm the column headings.

of admission of coal. The impact and the slicing action produced by the hammers as the coal travels from near the point of hammer suspension to the hammer tips and from the point of admission to the large opening in the grid circle, results in the necessary degree of reduction in the first effect.

The method of introducing coal into the pulverizing zone of the swing hammers is believed to be unusual. In place of admitting coal at the periphery of the hammer tips, the coal is admitted from the side of the hammers and near their pivotal point. The entire length of the hammers is used for pulverizing as well as the tips. Wear of the hammer edges is evidence of this action.

The original conception of the swing hammers and grid was for provision against damage by foreign material, and to increase the reliability of the machine. It was subsequently found that the substitution of hammers and grid resulted in a decided reduction in power.

The swing-hammer arrangement apparently is the most effective means for coarse pulverizing, and the attrition arrangement in the second effect the most efficient method for fine pulverizing. The combination resulted in a machine requiring approximately half the power of the original machine.

The essential characteristics of a unit pulverizer were stated above and the development of these characteristics together with other characteristics is discussed below.

Reliability. The design developed permits foreign material to pass into the machine without injury. The swing hammers throw the metal out of the opening in the grid circle. The metal lodges in a pocket at the bottom of housing and may be removed by opening a slide.

Ability to Pulverize Wet Coal. The present machine has the same ability to pulverize wet coal as the original machine. It is a combination drying and pulverizing machine, especially when hot air is admitted to the first effect.

Power Consumption. Curves for conditions G and J indicate the reduction in power consumption and the final power consumption obtained. At 4000 lb. of coal per hour, the kilowatt-hours per ton at the Atritor shaft was reduced from 18.3 to 9.2. At 1500 lb. per hour, the reduction was from 27.4 to 13.6. The assembly of curves in Fig. 9 indicates the effect of the successive changes on the power consumption of the machine.

Maintenance. The pressed-steel ferrules have permitted a wide latitude in the shape of the pegs. The hard cast metal used has shown a greater resistance to abrasion than any other metal. The combination of shape and material has resulted in a decidedly longer life of pegs and economical maintenance.

Fineness of Grinding. The present machine produces an average fineness of 90 per cent through 200-mesh with Eastern bituminous coal having 7 per cent moisture. Practically all the coal except a trace passes a 50-mesh screen. The proportion of extreme fines which are indicated by the microscope but which cannot be accurately determined by screen tests is quite large. The existence of such fines in pulverized coal produced by certain classes of pulverizers and the lack of them in pulverized coal produced by other classes, is shown in a paper entitled A Microscopic Study of Pulverized Coal, by L. V. Andrews, published in the May, 1925, issue of MECHANICAL ENGINEERING, p. 429. This paper also points out the desirability of such fines.

Omission of rejector blades reduces the fineness from 10 per cent to 15 per cent, but the power is also reduced by about 2 kw-hr. per ton at 4000 lb. per hour. The present machine has the same characteristics as the original machine in being able to produce pulverization which is independent of wear of pulverizing elements. The degree of fineness is substantially the same after 1000 tons have been pulverized as when all parts are new.

Capacity. Capacity tests have been made with Eastern bituminous coal having 3 to 5 per cent moisture at 6750 lb. of coal per hour, the actual energy absorbed by the pulverizer at this rate being 6.7 kw-hr. per ton. The fineness was 81 per cent through 200-mesh with 2.6 per cent remaining on a 50-mesh screen.

An effort was made by the authors to procure technical and scientific information relating to pulverization of material, but they found that such information was meager, and also that it was more or less empirical. They have endeavored to apply a rational process to the development of this particular problem, with the results as indicated.

It has usually been assumed that an approximate irreducible minimum had been reached in power consumption for pulverization. A more scientific knowledge of the actual processes and forces involved in the reduction of particles may prove that this is not true. It is hoped that this effort to record pulverization phenomena may stimulate more research along this line, and thus advance the art.

Manufacture of Commercial Steel Helical Springs

Present Status of the Art of Manufacturing Small- and Medium-Diameter Springs—Trade Requirements—Methods of Manufacture—Materials and Their Selection

By F. H. BROWN,[1] WORCESTER, MASS.

THE manufacture of commercial springs, like that of all other mechanical devices, has progressed from a crude process using simple tools to a modern system in which automatic machinery is employed. This advance has naturally followed along with that of machine tools and other machinery used in the shoe, textile, paper, agricultural-implement, and automotive industries.

There are innumerable forms into which spring material can be made in order to function as a spring. However, in this paper the author will consider only the most common one, namely, the helical spring, of which there are two general classes—compression and extension.

Trade Requirements

Until a few years ago many machine designers made no calculations of the springs required in their mechanisms, depending upon the spring manufacturer to supply a suitable spring either from a meager description of the mechanism or possibly from a visual examination of it. Sometimes designers specified dimensions suggested by their sense of proportionality or derived from such calculations as they could make using formulas found in handbooks—and often the wrong formula.

Thus there are many machines today operating with springs so designed, and unfortunately a large number of them cannot be re-equipped with correctly proportioned ones due to space limitations or the necessity of making costly changes on patterns or forgings.

The manufacturer therefore has inherited, so to speak, a considerable volume of business which will always be a constant source of complaint as long as the machines in which these springs work are in existence. This condition is not entirely regrettable because it is believed to have been responsible for two forward movements: first, the urge to manufacturers to improve the quality of materials and processes; and second, the awakening of designers to the fact that springs, too, must have a factor of safety as well as bridges and other structures.

Today we are confronted with an era of exacting specifications. Too many tolerances are specified without knowledge as to their relation to manufacture. In the past a manufacturer made mental reservations on such absurd tolerances, but today this is not "good business." Inspection departments have no authority to accept springs not technically agreeing with the blueprint, despite the fact that for all practical purposes the springs will function properly. Thus, when specifications with unreasonable tolerances are received by the company with which the author is connected, the customer is advised as to what tolerances would ordinarily be expected from its wide experience on such a spring. Unfortunately, a manufacturer's honesty is not always respected in this matter, and answers are often received informing the company that the customers cannot allow the values suggested, and that they have never had any difficulty in purchasing to such specification for many years. Our repeated investigations of such cases invariably discloses that customers have not thoroughly examined previous shipments, that they have no suitable measuring devices to warrant such inspection as these tolerances require, and that the specifications are not thoroughly detailed to insure comparative results.

In the future we are expecting that engineers designing springs will consider more fully the conditions of spring manufacture. It is only by working close to the manufacturer that specifications can be intelligently drawn up. We believe that future specifications will give more information as to the spaces in which and conditions under which the springs work, making tolerances only where necessary and as large as the spaces will permit. No reputable spring manufacturer will purposely use any greater tolerances than good commercial practice requires; and it should be the aim of progressive engineers to design spaces wherever possible to allow for commercial springs, if for no other reason than to secure reasonable costs.

Methods of Manufacture

Springs, from the manufacturing standpoint, are of three kinds: commercial, selected, and experimental. Manufacturers of springs are chiefly concerned with the first two as they comprise the volume of their profitable business. This paper will not deal with experimental or scientific springs, inasmuch as this field has only been surveyed for future exploration. However, it is a fertile one for investigation.

Commercial Springs. The majority of orders received for springs result from inquiries for specified quantities of springs to be made in accordance with samples or blueprints. Based upon a given sample or blueprint, a mill record is made showing the complete dimensions, the material to be used, and the individual operations through which the material must pass. This is all decided by the engineering and cost departments before the order is executed.

A wire manufacturer making springs naturally has available many kinds and grades of spring materials and complete information as to their chemical and physical properties, and unless the blueprint calls for a definite material he will use that which in his opinion is most suitable considering all factors, including intended use. In the case of a sample the material is analyzed and duplicated wherever possible, unless from other conditions specified the author's company feels obliged to recommend a more suitable one. For instance, a sample may be received made of a wire having a low elastic-limit value and which takes set due to design. Here a better grade is substituted in order to meet the design.

Before considering the various grades of steel which are used in springs, the general commercial practice of manufacture will be outlined.

Small Orders. It does not pay to set up automatic equipment to manufacture in small quantities, and consequently springs are made by the simple process of winding wire on an arbor placed in a lathe, a suitable guide being provided for winding the coils either close or open. There are two important things to remember in conjunction with design when springs are made by this process: First, the wire stretches and the loads the springs will carry are lighter than in the case of those made automatically, in which the wire upsets; second, in tension springs thus wound the initial tension available is less than in machine-made springs, especially those with large indices, since the wire cannot be made to lag without riding up on the coil.

After winding, the coil of wire is taken from the lathe and cut into the proper lengths prior to finishing the ends. The ends of compression springs (see Fig. 1) are finished plain or squared as desired. Either kind of end can be, and the latter usually is, ground by hand in small quantities and in automatic double-end grinders in large quantities. Springs wound on low stresses are given a low-temperature heat treatment to set them for lateral stability. Springs of high stresses are given mechanical work in conjunction with this low-temperature heat treatment to develop torsional elasticity approaching that of the original tensional value.

The ends of tension springs (see Fig. 2) are furnished plain for use with screw plugs, or with hooks or loops. Others have special fastenings such as separate hooks and bolts. It is quite evident that special tools are required for all except the regular styles, and in small quantities such ends must be made by hand with appliances not especially adapted for uniformity.

The springs are then ready for finishing, and may be left plain or rattled, electroplated, color-blued, or japanned. By far the majority of springs are japanned. "Plain finish" means the original

[1] In charge of Spring Factory, American Steel & Wire Co.

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finish of the wire as processed. The springs are then ready for inspection and shipment.

Large Orders. To make a thousand, twenty-five thousand, or a million springs by the process just described would be tedious, so machines are employed known as automatic coilers and hookers. In these the wire is drawn by feed rolls from the reel and pushed through suitable guides to a stationary arbor. Here winding tools actuated by cams wind and cut springs automatically in rapid succession. The ends of the springs can be squared or the diameter and pitch varied at will by employing the proper design of cam.

Extension springs are also made on such coilers and suitable ends provided in a separate operation. If the quantity permits and the

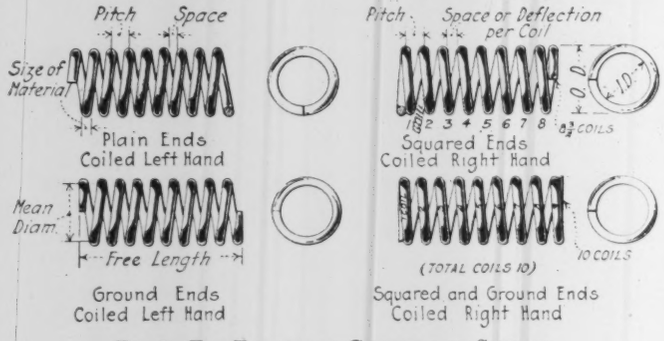

FIG. 1 END FINISHES OF COMPRESSION SPRINGS

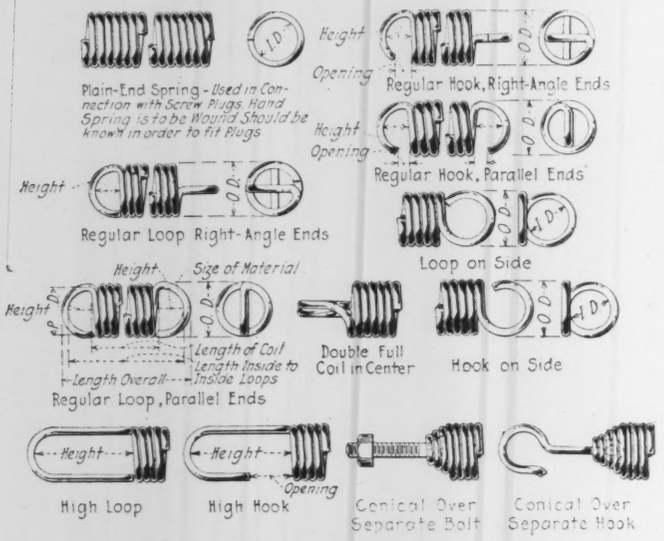

FIG. 2 END FINISHES OF TENSION SPRINGS

hooks are regular (approximately one-half of the end coil turned up across the center of the spring), this can also be accomplished entirely automatically.

The maximum amount of initial tension possible to wind into a spring is a load equivalent to a stress of 35,000 lb. per sq. in.; 15,000 to 25,000 lb. per sq. in. is normal.

Automatically wound springs are treated, depending upon stresses and material, the same as hand-made springs. On such springs, however, it is always best to use the proper design and material to avoid additional hand labor, as this has a marked effect upon the costs of springs originally produced at so low a cost.

Control of Variables. It is needless to say that proper control of operations requires preliminary testing where specifications are to be met. It is a fact that the additive effect of all the variables encountered in spring manufacture is so great that in order to consistently meet complete specifications every lot of springs must be tested by making preliminary trials. The commercial tolerances of wire and rod manufacture have a great bearing on the loads. For instance, specifications and records might call for No. 12 wire (0.105 in. diam.). The wire received at one time might run from 0.103 in. to 0.105 in., while the next lot might range from 0.105 in. to 0.107 in. As the load varies with the fourth power, it can readily be seen that unless this irregularity of size is counteracted by more coils or larger diameter or less free length, the starting standard is wrong and the regular variations in manufacture will bring many of the springs outside of the load limits which ordinarily would be considered reasonable. Therefore as much leeway in dimensions as possible should be allowed the manufacturer to compensate for load where load is an important factor.

Springs that are made to a given standard are partially inspected prior to shipment to insure that the majority meet the specifications. A complete laboratory with special instruments is set aside for this work.

Selected Springs. Many mechanisms are designed where the spring is the heart of the machine. The failure of the spring to do its work within a predetermined range causes the mechanism to fail, and usually great damage is done to other parts. In such cases the spring has a very important role, and any reasonable cost to produce it should be warranted.

There are various ways of producing selected springs, the first of which is to make them commercially and by inspection discard those outside of the tolerances agreed upon for acceptable springs. This is an uncertain method, and if the tolerances are very close, reasonable delivery is out of the question.

The surest way is to select the wire for size and temper, and to employ additional hand-inspection operations to insure a large percentage of the product's passing the final 100 per cent inspection. An example of this would be reaming the ends for inside diameter and working each individual spring to a uniform free length before setting the strains of coiling with low-temperature heat treatment.

MATERIALS

"Steel spring wire" is not a definite term and is borne by many grades of wire. With the exception of large hot-wound springs and special cases of scale springs, however, it can generally be said that the majority of springs are made by cold-winding wire of relatively high elastic-limit value.

Classifications. For springs using from 0.003-in. to 0.156-in. wire, music spring wire, often termed "piano wire," is frequently used, especially in the smaller sizes which are drawn wet. It should be remembered, however, that music spring wire is not a definite material as is the case with piano wire, which is actually made with definite physical characteristics for acoustic purposes.

Music spring wire has approximately the same chemical analysis as piano wire, but is made in varying tensile strengths and by different processes of drawing and "patenting." This materially affects the toughness of the wire, and, in the author's opinion, the higher the strain in the wire, the shorter the life that can be expected from the spring. In order words, springs that are vibrated rapidly should not be made of wire with maximum elastic-limit values but of wire drawn by slow and small-percentage drafting. This naturally increases the cost, but serves to show what is done in acute cases where springs are used under high velocities, even though only stressed to 60,000 lb. per sq. in.

Music spring wire when drawn to a high strain has the highest elastic-limit values of our common spring materials, but the value in tension is much higher than that obtainable in torsion. The greater the elastic-limit value in tension, the more responsive is the wire to high torsional values. However, there are other physical limiting values to be considered. In "surging" or compressing a spring wound on a very open pitch to the solid height for the purpose of putting in cold torsional work, it is found that springs of small indices (index = mean diameter of helix divided by size of wire) approach close to the elastic limit of the material in tension.

Thus in small gun springs made of 0.030-in. music wire with an index of 3, we are able to secure as high as 200,000 lb. per sq. in. torsional stress. The same wire wound to an index of 20 could not, due to mechanical difficulties, be wound on a sufficiently open pitch to cause as much cold work to be done upon the spring when surging it.

The next spring wire of commercial importance, and the greatest from a tonnage point of view, is called "oil-tempered wire." This is most commonly used in sizes of from 0.031 in. to 0.500 in., although as large as $\frac{5}{8}$ in. has been made. This is a 0.60–0.75 per cent carbon steel, either acid or basic open-hearth. It is hardened in oil and tempered in lead in a continuous process.

The very small sizes can be furnished with a polished finish or bright-tempered. The larger sizes are always black-tempered,

just as the wire comes from the furnace. The tensile strength of this wire ranges from 150,000 to 250,000 lb. per sq. in., the lower values for the larger sizes. The elastic-limit value in tension is approximately 70 to 75 per cent of the tensile strength as compared to 50 to 65 per cent in the case of cold-drawn wires such as music wire. With oil-tempered wire and even with small indices it is more difficult to approach the elastic limit in tension by surging than with music wire.

Premier or Crown spring wire is used in large quantities where a cheaper material is required than either music or oil-tempered wire. It is usually a 0.50–0.70 per cent carbon steel with a relatively high manganese content. It obtains its tensile strength through patenting and drawing the same as music wire, but naturally by a more rapid process. It is commonly made and used in sizes ranging from 0.03 in. to 0.250 in., although the larger sizes of springs are today being made from oil-tempered wire owing to the higher elastic-limit values obtained from this grade of material.

Low-carbon bessemer wire is occasionally used in common products where the stresses are low or where the springaction is secondary to the forming.

Special alloy steels are used for springs, which can be made from annealed wire and hardened after forming, or from oil-tempered wire or from cold-drawn wire. Those that have been used are chrome-vanadium, silico-manganese, and chrome-molybdenum.

Under certain conditions reports indicate that longer life can be secured from these steels for helical springs than from carbon steels. The experience of the author's company, as far as helical springs are concerned, is that carbon steels give in practical use superior service to alloy steels in most cases. Owing to the wide variations in life-test results between any two springs of the same material—due possibly to unequal conditions of manufacturing and test—it is believed that life testing is yet to be developed to a point where it will be indicative of the quality of the material.

Selection of Materials. A wire manufacturer making springs can use many grades of cold-drawn spring wires which are not usually marketed under standard practice. The individual-design and spring-manufacturing conditions govern the wire manufacturer in making such wires. A knowledge of a few of these conditions will aid in understanding this statement.

Polished music wire, for example, has a very smooth finish, and if a spring made of it is of small index the power required to push it into an automatic coiler will be so great that the wire will slip in the rolls, at times causing a large variation in the number of coils. To overcome this, special dry-drawn wire is made, which has a dull coppered surface but equal physical properties.

Again, cold-drawn wires, particularly music wires, have a cast in the bundle which causes a great variation in pitch on springs automatically coiled. This is especially true regarding springs of large indices and close pitch. Here the manufacturer must be sure to use the lowest-strain wire possible and still meet the stresses, as high-strain material in any grade is liable to have a heavy cast.

As soft a wire as possible should be used where vibration and shock are excessive. This must be done in designing, however, as stresses must be low to withstand the forming process. In cases where music wire is necessary, due to fairly high stresses the wire manufacturer can use special-process material which has been given light and slow draftings.

Summary. It is seen, then, that there are many available grades of materials and that each can be varied to suit conditions, provided these are known in advance and the cost of the finished article warrants the special costs which attend deviations from standard practice. Very large quantities are usually necessary to secure deviations, as small lots of material do not warrant the changes in settings in the rod or wire mills or in the spring mills.

The author realizes that any paper on so extensive a subject as the art of manufacturing springs can only hope to deal with a few of the details encountered in problems of production and design. He trusts that the foregoing discussion will throw new light upon the matter, and that hereafter designers will not consider the adoption of a spring design without giving due consideration as to how it is to be made.

This is an age of specialists, and spring manufacturers are no exception to the rule. After deciding upon what it is desired to accomplish, make a tentative design and submit it for comments to a reputable spring manufacturer, with complete information as to what the spring must do and where it works. With his comments at hand a final decision can be reached as to the specification, and in most cases the spring problem can then be forgotten.

Characteristics of Weighing Springs

By J. W. ROCKEFELLER, JR.,[1] NEW YORK, N. Y.

IN DEALING with the subject of weighing springs, it would be well at the start to define the term. In scale construction many types of springs are used. Indeed, no scale can be truly said to be springless, as the placing of a load on the scale platform produces stress—and consequently strain—in all members supporting the load. This stress may be in the nature of straight tension if the member affected is a steel tape; it may be flexural, as in a spring-plate fulcrum or cantilever auxiliary spring; or if a helical spring is used in the scales construction, it will be torsional. In any case the member is distorted by the load and returns to its original position upon removal of the load.

Weighing machines in general may be classified according to the amount of elasticity they possess, as follows:

1 Those in which the elasticity is incidental to the construction as scale beams, equal-arm balances, some pendulum types, etc.

2 Those in which a spring is used to indicate whether or not the scales are in balance and to measure the degree to which they are out of balance within restricted limits, but in which the spring is not used to measure the magnitude of more than a small part of the load. Into this class would fall some types of scales commonly called "over and under," as well as scales embodying the plate-fulcrum construction.

3 Those in which the spring is used to measure the entire load or a large part of it.

The third class includes the majority of indicating scales used commercially. These scales are commonly referred to as "spring scales" or "spring balances," and it is the springs used in these instruments that will be designated hereinafter as "weighing springs."

In a commercial scale the spring has two very important duties to perform: During the life of the instrument it must receive and absorb many shocks so that their effect upon the knife edges and other delicate parts will be minimized. In this way the springs in a scale perform the same duty as those in a motor car, locomotive, or other highly refined mechanism.

The second duty performed by the spring is that of indicating the weight of a body by the amount the spring deflects under the load. In performing this second duty, the most necessary characteristic of the spring is that of giving a straight-line load-deflection curve. With the deflection proportional to the load imposed, the scales dial may be uniformly calibrated (i.e., the divisions may be equal) without resorting to cams or other corrective devices to bring about this condition.

It is well known that helical springs obey the law of proportionality discovered by Robert Hooke, and if all initial tension is removed from the spring (as is done in the case of weighing springs) a highly accurate commercial balance of simple construction may be manufactured.

The absence of initial tension is the most important characteristic embodied in scale springs, and the main one distinguishing them from other types of extension springs.

The effect of temperature changes on the modulus of elasticity of the spring is a factor that merits attention when scales are subjected to extreme temperature variations. This effect may be overcome by a device similar to that used in time-measuring instruments or by using for the spring a wire of an alloy the modulus of which is not affected by temperature changes.

In commercial balances a weighing is usually made in less than five minutes, after which the load is removed and the springs are under no tension except the very slight tension caused by the weight of the platform. If the springs used in such instruments are properly made, they will retain their accuracy indefinitely. It is not uncommon for a spring scale to be in service for twenty years or more, receiving no attention, and to perform as accurately at the end of this time as when it was shipped from the factory.

In recent years the spring balance has been used more and more in the field of engineering. In testing machines the spring, due to its small inertia, has come into common use. In scientific work where it is necessary actually to measure weight, a spring instrument is the only type of scale that can be used.

In such work a new factor comes into prominence as affecting weighing springs, and this is the part that time plays in the deflection caused by the load.

When a spring is highly stressed by a constant force it will be found that its immediate deflection will be of one magnitude, while after twenty-four hours it will be slightly greater, and after another twenty-four hours, still greater yet under the same load. However, these increments of increasing deflection diminish constantly, the time-deflection curve approaching an asymptote at a deflection value which may be calculated from a value known as the "true modulus" or "limiting modulus" of elasticity.

It has been found that as the fiber stress decreases, this limiting modulus approaches the common or immediate modulus, and the amount of drift, hysteresis, or secondary elastic deformation, as this latter deflection may be termed, can be controlled by properly designing the spring until the error produced by this retarded deflection comes well within the other errors of the balance. In this way the problem presented is solved from a commercial aspect.

As a subject for research, however, the matter of retarded deflection presents many interesting phases. No satisfactory explanation has yet been given for this time effect on the stress-strain diagram of materials. It is altogether likely that the most plausible theory will take into account the fact that when an elastic body is placed under tensile, compressive, or torsional stress, the rate of motion will be governed by the unbalanced forces acting upon the material. As the amount of unbalance force is decreased, the rate of motion will be decreased.

TABLE 1 INCREASE OF DEFLECTION UNDER CONTINUED STRESS

Spring No.	Number of hours under stress: 24	48	500	1500	4000	8000	Stress lb.
1	0.0 (0.0)	0.0 (0.0)	0.0 (0.0)	0.0 (0.0)	0.0 (0.0)	0.0 (0.0)	5,000
2	0.0 (0.0)	0.0 (0.0)	0.0 (0.0)	0.0 (0.0)	0.0 (0.0)	0.0 (0.0)	10,000
3	0.0312 (0.0312)	0.0312 (0.0)	0.0312 (0.0)	0.0312 (0.0)	0.0625 (0.0312)	0.0625 (0.0)	20,000
4	0.0312 (0.0312)	0.0312 (0.0)	0.0625 (0.0312)	0.0625 (0.0)	0.0937 (0.0312)	0.0937 (0.0)	30,000
5	0.0312 (0.0312)	0.0312 (0.0)	0.0625 (0.0312)	0.0937 (0.0312)	0.1250 (0.0312)	0.1250 (0.0)	40,000
6	0.0312 (0.0312)	0.0312 (0.0)	0.0312 (0.0)	0.0625 (0.0312)	0.0937 (0.0312)	0.0937 (0.0)	50,000
7	0.0312 (0.0312)	0.0625 (0.0312)	0.0625 (0.0)	0.0937 (0.0312)	0.1250 (0.0312)	0.1250 (0.0)	60,000
8	0.0625 (0.0625)	0.0937 (0.0312)	0.0937 (0.0)	0.1250 (0.0312)	0.1875 (0.0625)	0.1875 (0.0)	70,000

Table 1 gives the results obtained on some very rough preliminary work dealing with this problem. The actual measurements made were in fractions of an inch, the limit of precision in measuring being in the neighborhood of $1/64$ in. They are reduced to decimals in the tables merely for purposes of comparison.

The nature of the tests was such as to permit of nothing but the broadest generalization in interpreting the table. It may be seen, however, that for low stresses the amount of drift (as shown by the values in parentheses in the table) is very small, while for stresses of greater magnitude it is correspondingly higher.

It will also be seen that the greatest amount of drift takes place in the first few hours, and that after 4000 hours it is in no case discernible. Tests now in progress confirm these assumptions. It is recommended that the study of this secondary deflection be made a subject for research. Its bearing upon springs for weighing instruments is important, and its application to the general laws of strength of materials, direct.

[1] Works Manager, John Chatillon & Sons. Jun. A.S.M.E.

Contributed by the Special Research Committee on Metal Springs for presentation at the Annual Meeting, New York, November 30 to December 4, 1925, of THE AMERICAN SOCIETY OF MECHANICAL ENGINEERS. All papers are subject to revision.

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Springs for Electrical Measuring Instruments

By B. W. ST. CLAIR,[1] WEST LYNN, MASS.

THE springs of electrical indicating instruments operate under unusual circumstances. The mechanical conditions are quite unique and very severe. They are not the only consideration, however, as the electrical properties of the springs are of great importance.

The mechanical requirements of the instrument spring can be stated as follows:

1 The spring shall have a constant torque for a given deflection irrespective of the temperature or the time of application of stress

2 The zero position shall be stable and independent of temperature or the previous stress history of the spring

3 The spring should be mechanically strong enough to retain shape and zero position during shipment.

The electrical requirements are:

1 The specific resistance of the spring material should be as low as possible

2 The temperature coefficient of resistance should be as near zero as possible

3 The spring should be of non-magnetic material.

Electrical indicating instruments are capable of great accuracy, greater perhaps than any other common measuring device whose calibration depends upon springs. On a scale 90 deg. long it is not unusual to expect an accuracy of 0.1 per cent, and in some instruments of long pointer length it is usual to expect the working error to be less than 0.05 per cent. In other words, it is expected that the overall working errors will be less than 5 minutes of arc and 2.5 minutes of arc, respectively. These errors include many factors other than springs, so that the actual spring error usually must be even less than these figures.

The materials that best meet the mechanical and electrical requirements are of a bronze nature, generally of a phosphor-bronze type. The exact composition of the bronze is of very great importance. Materials of a ferrous nature are inapplicable because of their magnetic properties and their proneness to oxidation.

The shape of the spring and the technic of its manufacture are of great importance. They are almost universally of spiral form

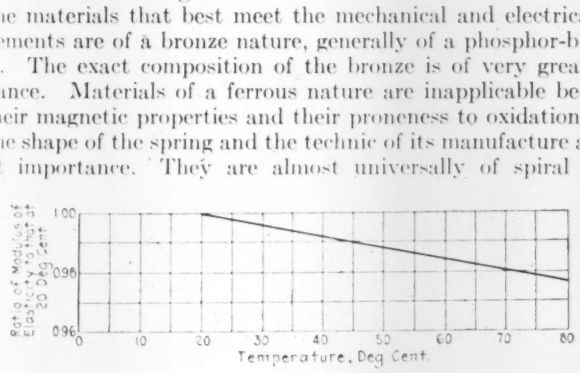

FIG. 1 EFFECT OF TEMPERATURE ON YOUNG'S MODULUS OF ELASTICITY OF PHOSPHOR BRONZE

and of relatively few turns. The methods of reducing the usual round wire to a ribbon and the circumstances of tempering the wound flat ribbon are of extreme importance. So also are the methods of attaching the spring to its abutments in the instrument.

A very common source of error is that due to the temperature coefficient of elasticity of the spring. While this coefficient is small, it is of sufficient magnitude to cause serious concern to the instrument engineer, who must build instruments that show no appreciable temperature change. Fortunately, in some cases this change just matches and nullifies other changes in other materials. This is true in direct-current instruments that depend upon permanent magnets. The magnets decrease in strength with temperature, and so do the springs. The net change from these two causes is very small, indeed. Fig. 1 shows the variation of the coefficient of Young's modulus of elasticity with the temperature.

Instruments are used under many adverse conditions, some being employed for only momentary readings, while others are left permanently in circuit. The springs as well as other parts must show no change under load, with the time, and should show no time lag of any sort.

A source of great concern to the instrument designer and user is the effect in springs that has been termed "hysteresis," and, less correctly, "fatigue." This is manifested by a slow return to zero from a position slightly above zero after a more or less extended period of stress. Two other phenomena have at times been associated with hysteresis, one a permanent change of zero after a period of stress, to which the term "set" has been given. The other is a thermostatic type of phenomenon, as it is manifested by a change of zero with a change in the spring temperature. Both these latter, however, are now relics of the past, as they do not appear in any of the well-designed springs of today. Hysteresis itself, though, is

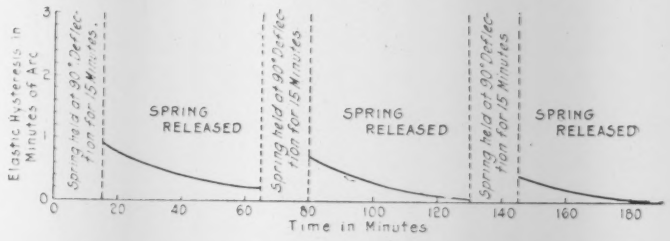

FIG. 2 RELATIONSHIP OF ELASTIC HYSTERESIS IN INSTRUMENT SPRINGS TO PROLONGED REPEATED STRESS

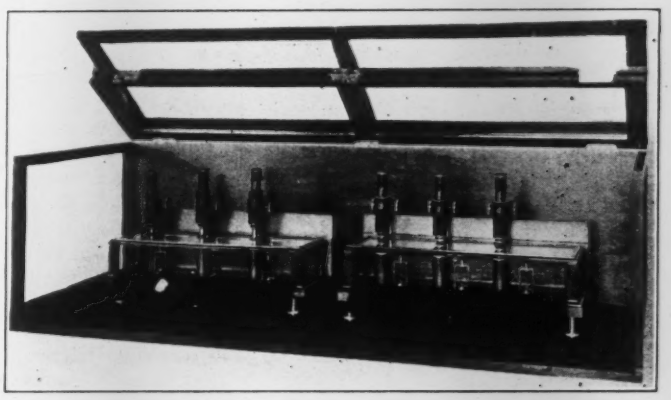

FIG. 3 MICROSCOPE COMPARATOR FOR DETERMINING HYSTERESIS AND OTHER ZERO UNCERTAINTIES

one of the chief concerns of the designer. It is determined by the purity and composition of the material, and especially by the methods of working, by the hardness, forming temperatures, etc. The proper recognition and control of these many factors has led to the development of springs whose hysteretic changes are very small, as seen in Fig. 2. The initial hysteresis after twenty minutes' stress at 90 deg. is less than 1 minute of arc. Longer processes and more attention to detail permit the commercial production of springs with hysteresis one-quarter of the values shown. Fig. 3 shows a type of microscope comparator for the determination of hysteresis and other zero uncertainties.

Unfortunately, the mechanical properties are not entirely the ruling ones in many instruments, as the electrical ones are of prime importance. The springs in many instruments must also serve as flexible conductors to connect the armature windings with the external portions of the circuit. The current that can be carried by the springs without appreciable rise of temperature represents one limitation. Another is the temperature change of resistance of the spring. This is a very serious factor in direct-current millivoltmeters and milliammeters. The total resistance of the instrument is small. The temperature coefficient of the completed instrument is practically the coefficient of resistance of the circuit. The springs representing a large part of the total resistance, are

[1] Standardizing Laboratory, General Electric Co.

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FIG. 4 LOCATION OF COMPENSATING WINDINGS WITH RESPECT TO INSTRUMENT SPRING

hence a vital factor. In many cases it is necessary to sacrifice very desirable mechanical properties to obtain high electrical conductivity. In direct-current voltmeters, where the circuit resistance is several hundred or thousand ohms, this is not a serious factor. The ratio of copper winding and spring resistance to the total circuit is extremely small. It is not difficult to make instruments of this class whose temperature change in indication is less than 0.01 per cent per deg. cent.

In certain alternating-current instruments where the power loss is quite large, special provision is necessary to counterbalance the change in indication as the instrument heats up from standing in circuit. The changes in indication are due to several factors, but in certain cases the ruling one is the weakening of the springs with rising temperature. In some cases it is necessary to include windings of copper placed as near the spring as possible, so as to have as closely as possible the same rate of temperature rise as the spring. Fig. 4 shows the location of the copper compensating windings with respect to the spring in one instrument.

The range of full-scale torque of indicating instruments is quite large, but all the torques are really quite small. The usual range is from 0.1 gram-millimeter to 10 gram-millimeters. These figures when translated into the more usual units are startlingly small—the torque range being from 1.0×10^{-6} to 1.0×10^{-4} foot-pounds. When it is remembered that these are full-scale torques and that the permissible error brings these figures to 1.0×10^{-9} to 1.0×10^{-7} foot-pounds as the required torque response, some of the unusual requirements of instrument springs can be realized.

The foregoing is but a brief statement of the general spring problem in so far as it relates to electrical instruments. No attempt has been made to discuss the interrelationship of the many factors that control the final properties of the spring. It is hoped that some of these may form the basis of future communications.

Influence of Plant Design on Plant Efficiency

Economical Selection of an Industrial Site—Basic Data Which Determine the Plant Design—Site Limitations Which Affect the Plant Layout—Design of the Plant

By HAROLD T. MOORE,[1] PHILADELPHIA, PA.

WHEN we observe a familiar article of commerce which is a recognized leader in its class, we sometimes wonder whether the company behind the product acquired its supremacy through a streak of luck or the vision of a genius. If we take time to ponder the question and have some knowledge of the results achieved, we may assign some of the popular reasons usually given for commercial success, such as exclusive patent rights, limited competition, national advertising, economic production methods, efficient plant layout, strategic location, skilled management, or able leadership. Whether we assign any or all of these reasons, we can generally conclude that the real success factors were constructive imagination, ability, and industry.

Whether the product made, the methods used, or the standard adopted are the work of one individual or the composite achievement of an organization, the visualizing in advance of the objective to be accomplished and anticipating the details to be perfected are obviously necessary before practical results can be realized. In all lines of endeavor, *imagination* is the advance agent of progress, without which our complex civilization could not function.

When applied to the industries, imagination furnishes the incentive which evolves the product, determined its quality, and suggests the initial methods of manufacture. When the sequence of operations, the type and capacity of the production facilities, and the desired output are known, the size of the plant needed can be readily determined and its arrangement designed to fit the process, provided, of course, that imagination and foresight are intelligently applied. It is this phase of plant design and the influence of advances planning on production efficiency that will be discussed here.

The majority of enterprises start on a small scale either in an existing industrial building where floor space can be leased, or in a small plant built or rented for the purpose. The large industries are mostly gradual developments from the small factories which have expanded by progressive stages, either at their original sites or at new locations. Some of the major industries are the result of consolidations or of new departments added to broaden the line or enlarge the manufacturing facilities.

At some stage in their development most plants require enlargement or partial rearrangement to handle the normal increase in business. Sometimes changes in the art or the market justify relocation to obtain more economic advantages or avoid burdensome handicaps. Whatever may be the reason for factory enlargement or new plant development, it is the opportune time for the management to have the needs of its process carefully analyzed and the plant designed to fit the process.

How Building Restrictions Handicap Manufacturing

Many factories have modern equipment and efficient methods but an unbalanced distribution of floor space, with some departments badly congested and others misplaced. Some industries located in multiple-story buildings are handicapped by a lack of flexibility, which might have been avoided if space had been available for single-story construction, or for a combination of single- and multiple-story. Some building widths or the spacing of columns are not well suited to the machinery size or spacing and prevent an ideal arrangement from being adopted. In some instances headroom is not sufficient for overhead handling by traveling cranes or trolley hoists.

Many of these handicaps might have been avoided if more care had been exercised in the selection of the site, the location of the original buildings on the property, and the design of the initial plant. Limited capital is frequently responsible for site or building restrictions, but often where ample funds are available, the locating and planning of the new plant are not always given the attention deserved.

Economical Selection of an Industrial Site

The factors affecting plant location are quite well known, the leading ones including:

1. Sources for obtaining raw materials, semi-finished or finished parts
2. Markets for the products
3. Transportation service and rates
4. Labor market—skilled and unskilled, men and women
5. Prevailing wage rates and attitude of labor
6. Cost of power and fuel
7. Municipal advantages
8. Housing facilities
9. Convenience of location
10. Cost of real estate
11. Advertising value
12. Possibilities for waste disposal.

Comparative Advantages of Alternative Sites

A possible site may be of ample size to be practically free from boundary restrictions, but it is of considerable assistance both in making a selection as well as in locating the first buildings on the property to have a preliminary idea of the type of layout preferred. A tentative plan of an ideal plant based on the floor space and process requirements without regard to property limitations is a safer check on the suitability of a site than only an estimate of the total floor space required. After the property is acquired it is quite likely that a modification of the ideal layout will be necessary to adapt it to the site. The location of highways, railroad sidings, residential districts, sewers and water lines, the points of the compass, the topography of the land, etc., will require consideration before the final layout is adopted.

Basic Data Which Determine the Plant Design

Preparatory to starting the design of a new industrial plant, the industrial engineers, to whom the plant-development work has been assigned, if not already conversant with the manufacturing process, should become familiar with all phases of the production and assembly operation, including types and capacities of the equipment to be used. This is necessary in order that they may acquire a mental picture of the process to be housed and be able to interpret the plant needs in terms of total capacity, floor space, types of buildings, and arrangement of departments. By reviewing, analyzing, and verifying the program submitted by the plant management, the engineers responsible for the plant layout must be able not only to visualize the various production problems to be solved, but must be sufficiently familiar with manufacturing methods in general and building economics in particular to supplement and develop the management's preliminary specifications and convert them into the finished plant built around the process.

This preliminary study of the various factors to be considered satisfies the management that the expansion program as planned is the correct one, or that it will be after certain modifications. It likewise assures the engineers as to the correctness of the basic data acquired before they complete the plant-development work.

The major considerations which have a bearing on space requirements or plant arrangement and which should be carefully analyzed and correlated before starting the plant design, include the following:

1. The desired capacity of the initial plant and the estimated future capacity to be anticipated in terms of production units
2. The divisions of the manufacturing schedule to determine the number and variety of the sub-assembled or finished units to be produced

[1] Industrial Engineer with Day & Zimmermann, Inc. Assoc. A.S.M.E.
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3 A list of materials or parts comprising the product to determine which ones will be manufactured and which ones purchased and stored

4 The production equipment or plant facilities needed for the desired capacity of the initial plant, including any special provisions or structural features which will facilitate production

5 A study of the manufacturing and assembly operations necessary to produce a finished or sub-assembled unit to check the proper spacing of equipment

6 The time interval required between successive operations, if any, to check the need for and the location of intermediate storage space

7 The sequence of operations in manufacturing and assembly departments in order that departments and equipment shall be in logical and convenient relationship for the progressive flow of materials

8 The space requirements per department to house the production equipment and provide the space needed for aisles, storage, and auxiliary departments

9 A review of the various operations entering into the process to determine whether certain departments should be isolated from the standpoint of safety, comfort, noise, or special process needs

10 A summary of the floor-space needs of the initial plant, which areas can be proportionately increased for the different departments, based on an assumed future capacity after a certain period of years, thus providing an approximate basis for estimating the space requirements and developing a layout suitable for the ultimate plant development.

The foregoing factors include the essential data upon which the design of the initial plant should be based. They establish the floor areas needed, the size of departments, the sequence of operations, and consequently the routes for the flow of materials. With this information available, the engineers who have studied the manufacturing needs of a particular industry should be sufficiently familiar with the problems involved to undertake the development of the plant design.

Site Limitations Which Affect the Plant Layout

With the capacity of the proposed plant known and the departmental needs approximated in terms of floor space and production facilities, both for the initial and ultimate plant development, the design program is established if there are no further restrictions. The engineers may have acquired sufficient basic data to enable them to work up preliminary layouts which will meet the process requirements in general, but the ideal layout may not be a suitable one for the particular factory site acquired.

It is therefore advisable for the design engineers to inspect the acquired or proposed site and obtain a mental picture of the local conditions which will have a bearing on the plant arrangement and its location on the property. A contour map of the property with boundary dimensions will be needed, together with complete information as to the size, location, and elevation of the nearest sewers, water and gas mains, the location of railroad tracks and highways, electric lines, etc. The bearing value of the soil should be checked and the city or state regulations in regard to sanitary, safety, and welfare provisions reviewed, especially if the industry is one with objectionable or hazardous features.

Design of the Plant

The majority of engineers are familiar with the use of templets of machine tools for the layout of departments. The evolution of the plant layout into a finished plan is largely a similar development of "cut and try," with templets conforming to the department areas placed in proper relationship to adjacent departments as determined by the sequence of operations. A sufficient number of arrangements should be made and studied to determine which plan has the greatest merit. The one with the most advantages should be further developed to incorporate as far as possible any additional features of merit brought out in the other preliminary plans.

The floor-space requirements and the process needs will determine whether and to what extent the multiple-story should be given the preference over the single-story building, assuming, of course, that there are no limiting conditions at the site. Special features of building construction to provide suitable headroom, overhead handling, satisfactory ventilation, natural lighting, fireproof walls, etc., will be evident from a knowledge of the departmental requirements.

The plant design, if efficiently planned in the manner described, will provide for the operating management plant facilities that are properly balanced and suitably arranged to insure economy of operation, and, when expansion is necessary, to permit of the plant's being enlarged without unbalancing the layout. By working from the ultimate layout back to the initial requirements it is possible to provide a condensed layout of the required capacity needed for the first group of buildings erected. The arrangement should be sufficiently flexible, however, to permit of step-by-step expansion toward the goal laid down by the development plan adopted without disarranging the production balance.

The temporary utilization of one building or department for another purpose is sometimes justifiable in the initial plant. It is usually less expensive to relocate a department later if the equipment is portable than it is to invest in excess plant capacity and isolate a department which would have an initial handicap of long haulage of materials.

In the foregoing discussion it has been assumed that the new plant is to be erected at a new site acquired for the purpose. It is frequently necessary to erect additions to existing plants to provide for increased capacity. The floor-space distribution in the existing plant should be analyzed in similar manner and a routing diagram made to trace the flow of materials.

The same principles as previously discussed will apply only with more limitations to consider in order to utilize to best advantage the existing floor space and locate the new extensions so that the progressive flow of materials will be satisfactorily maintained when utilizing the old as well as the new buildings provided for the purpose.

Effect of Plant Design on Its Operating Efficiency

After the plant has been constructed, it becomes the duty of the plant management to install and operate the facilities provided for the purpose. When the plant is being designed, the layout engineers and operating management must coöperate to insure that all necessary facilities for handling, production, sanitation, and safety are included and will best meet the requirements.

By the intelligent use of advanced planning and of constructive imagination, the new or enlarged buildings should house an efficiently arranged process in a well-lighted and architecturally attractive plant. The operating management is the beneficiary of the advantages or deficiencies that may be present in the finished plant. A correct analysis of the process needs, interpreted in terms of plant design and operating efficiency, is therefore an important service which industrial engineering is rendering to production management.

Industrial-Furnace Efficiency

Heat Losses Encountered in Industrial-Furnace Operation—Dilution of Products of Combustion—Application of Waste-Heat Boilers to Industrial Furnaces—Incomplete Combustion—Furnace Design

By VICTOR J. AZBE,[1] ST. LOUIS, MO.

PROGRESS during the past few years in the fields of boiler-plant combustion and heat absorption has been remarkable indeed. Boiler overall efficiency has climbed at a surprising rate, and in exceptional cases efficiencies in excess of 90 per cent have been obtained during tests. Even in small plants operating engineers realize the importance of the underlying chemical and physical factors which govern boiler-room economy; and, while they do not all test the waste gases for composition and temperature, practically all know the importance of high CO_2, proper baffling, operation at correct rating, etc. It is safe to state that during the last ten years the efficiency of the average boiler has been increased from about 55 per cent to 65 per cent. In view of the fact that this corresponds to a saving of fuel of over 18 per cent, the progress has been truly remarkable.

In the industrial-furnace field conditions are not quite as satisfactory. Thermal efficiencies in many cases are very low—in some plants efficiencies as low as 10 per cent are obtained. The principles of heat absorption and consequently of proper furnace design are imperfectly understood. This is especially true in certain fields. Saving of fuel often is considered secondary to rapidity of process operation to such an extent that little or no study is devoted to fuel economy. As a result of these conditions the fuel loss is tremendous, and in many cases the preventable loss exceeds 50 per cent. A preventable loss of 25 per cent is quite common.

It has just been stated that operating efficiency is often considered paramount to fuel economy. Since operating efficiency in most cases is dependent upon furnace-performance efficiency, the fact that little attention is paid to the latter seriously affects the first.

The reason for the tremendous fuel losses occurring in the industrial-furnace field is that the operations are usually of a high-temperature character. Enameling, smelting, annealing, lime and cement making, refractory burning, glass melting, etc., are all conducted, as compared with steam making, at relatively high temperatures, so the loss becomes correspondingly greater. The loss is especially great in cases when the product cannot be progressively heated and cooled to obtain a countercurrent heat exchange between it and the products of combustion and air.

One of the reasons for inefficiency is that most industrial-furnace units are so small that, to those who are in control, proper attention to efficiency does not seem profitable. This is in most cases a faulty assumption, but is in some cases correct. For example, waste-heat boilers with few exceptions are not economically justified in any but the larger installations and where waste-gas temperatures exceed a certain minimum.

There are numerous combustion engineers or engineers so-called in the boiler field, but only few in the industrial-furnace field. There is a considerable opportunity for advancement for those entering this field, but before they will become of much value, they will have to study heat generation and absorption much more intensely than it is taught in most engineering schools today. In addition to the theoretical knowledge, they will have to acquire a rather thorough knowledge of the process in connection with which the furnaces are being used; otherwise they will be of little value, and probably even of negative value. The knowledge of the men in many of the fields using industrial furnaces is based upon an empirical, slowly built-up foundation that comprises little, if any, truly correct scientific information. To bring such men to employ modern methods requires diplomacy. A man who does not inspire confidence by virtue of the thorough knowledge he possesses of the various ramifications of a given process, will accomplish little.

That which appears to be theoretically the best is not always practically the best. In the boiler plant, for example, with proper equipment the coal of least cost per heat unit is the coal that ordinarily proves the best. In some industrial cases this is not so. For example, in the enamel-ware industry in certain fields, natural gas, oil, and coal are all available, the gas costing three times as much as coal and the oil twice as much as coal; still, taking everything into consideration, the gas is generally considered the best when available. The reason is that the slightest speck of soot or ash impurity upon such a large surface as that which a bath tub presents may cause a loss that would more than offset any saving possible by the use of lower-priced fuel. With coal it may be necessary to use a full-muffle furnace in which the heat transfer is considerably retarded; with oil a semi-muffle may be employed, while with natural gas the heat can be applied directly to the manufactured articles and circulation currents may be created at will with hardly any danger of contaminating the surface with impurities. But the very reason that the higher-cost fuel is burned makes it so much more profitable to use all reasonable methods for conserving the heat. Further, when this is done, methods of control are developed that reduce the chances of contamination so that, in the long run, a scientific study will result not only in fuel saving but quite probably in a better product, in increased output, in reduced operating labor, or in reduced upkeep.

The losses encountered in the industrial-furnace field are the same as in the boiler field: namely,

1 Loss due to dry products of combustion
2 Loss due to excess air
3 Loss due to radiation
4 Loss due to moisture in the fuel and steam used for atomizing at oil burners if such are used
5 Loss due to incomplete combustion
6 Loss due to moisture from combustion of hydrogen
7 Loss due to unburned combustible
8 Loss due to moisture in the air.

The order of these losses becomes somewhat different from that in the case of boilers. The loss due to radiation becomes greater and more important.

Radiation Loss

The heat generated and supplied to a furnace or kiln or oven can be divided into that of higher temperature elevation which is available for employment in the process, and that of lower temperature elevation which is not available. Assume that we are dealing with an enameling furnace having an operating temperature of 1800 deg. fahr. and burning producer gas. All of the heat lost by radiation from the muffle, producer, gas flues, and wall surface between and including the producer and muffle, would have been available in the process if not lost. The heat of less than 1800 deg. fahr. elevation, if lost, is relatively inconsequential. The magnitude of the loss of heat of high temperature elevation is very much more than the actual B.t.u. loss. Assuming that the furnace efficiency is 25 per cent, the loss by radiation will be four times the calculated figure. If the theoretical potential temperature elevation is 4600 deg. fahr. and the muffle temperature is 1800 deg. fahr., and if the heat loss by radiation is 15 per cent of the total heat, 10 per cent from high-temperature zones and 5 per cent from low-temperature zones, we obtain the following:

A possible efficiency of the theoretical furnace without loss due to radiation will be 62 per cent; 10 per cent loss of total heat by radiation will cut the efficiency down to 52 per cent, and the fuel loss will therefore not be 10 per cent but rather $(10 \div 0.62 =)$ 16 per cent. Next, let it be assumed that due to various causes such as excess air and water vapor the highest temperature potential is only 3200 deg. fahr. The furnace efficiency without radiation will be 31 per cent, but with a radiation loss of only 10 per cent of the total heat it will be reduced to 20.8 per cent, so the actual fuel loss will not be 10 per cent but rather 33 per cent of the total fuel burned.

[1] Consulting Engineer. Mem. A.S.M.E.

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The above makes it imperative that heat of high elevation on the thermometric scale be conserved as much as possible. Still it is most unusual for one to encounter a plant where the gas flues are insulated. Gas initially at a temperature of 1300 deg. fahr. is passed through long ducts lined as a rule with only $4\frac{1}{2}$ in. of brick. Not only should the flue, producer, and other radiating surfaces be insulated, but a special effort should be made to reduce their extent since radiation loss often is the greatest of losses in the industrial-furnace field.

In quite a number of plants where the installation is apparently quite modern, consisting of a gas producer and an elaborate distributing system for the gas, the combustion efficiency is less than in other plants where operating conditions are apparently poor. In several instances analyzed by the author it was found that this paradoxical situation was due to heat lost before the gas was delivered to the kilns. The gas as leaving the producer will contain in sensible form between 15 and 20 per cent of the heat originally in the coal. If all this heat were to be lost and the gas delivered cold to a furnace operating at 1800 deg. fahr. the available heat would be 33 per cent less than with hot gas, the efficiency being reduced from 60 to 40 per cent. This of course is somewhat exaggerated; normally, the gas would not be allowed to cool to this extent.

In some ways it is difficult to exaggerate fuel losses that occur in the industrial-furnace field. The author in his testing-work calculations seems never to catch up with the actual losses—they always are more than the figures require them to be.

It is in part due to radiation loss that fuel oil and natural gas are so desirable. The heat can be generated almost up against the product being heated, with chances for loss due to this cause greatly reduced.

Loss Due to Dry Products of Combustion

The loss due to the necessary dry products of combustion is not controllable to any great extent unless recuperators or regenerators are used. If the loss due to radiation is kept small and the gases are cooled down to only small terminal temperature differences in the furnace, then about the only other factor influencing is the selection of fuel. For example, the products of combustion of raw producer gas at 1800 deg. fahr. will contain 52 per cent of the total heat, while blue-water-gas products contain only 36 per cent. Of course, one fuel should not be advocated over another excepting for sound economic reasons. Yet in spite of some theoretically undesirable features, producer gas has certain advantages that will make its use more general in the future.

The simplest way in which to cut down the loss due to dry products of combustion is to use recuperators—or what might also be called waste-heat air preheaters. They are a decided advantage, although many engineering data regarding their performance are still lacking. In some industries their use is as yet almost unknown.

The fuel value of recuperators is in some industries limited by refractory-furnace-lining difficulties. In fact, the wear of firebrick is one of the most common excuses offered for furnace inefficiency. A boiler cannot be efficient unless the furnace temperatures are high; the same applies normally to an industrial furnace, only more so, due to the rather high working-temperature levels that are ordinarily employed.

Dilution of Products of Combustion

The loss due to the products of combustion could be looked upon almost as a necessary evil and would not be so serious were it not multiplied by unnecessary excess air and diluents, such as steam. If it is worth while to keep the excess air low in the case of boilers with their heat-absorbing level of about 370 deg. fahr., it certainly should be so when the working level exceeds 1000 deg. and probably 1500 deg. While 10 per cent of CO_2 from a boiler furnace ordinarily is not considered bad and may be even good, this same dilution may with some kinds of high-temperature furnaces be fatal and stop production. A lime kiln operated continuously with 9 per cent of CO_2 will not make sufficient lime to warrant its operation, and a lime kiln has a relatively low-temperature heat-absorbing level.

It is due mainly to this heat loss that gaseous and liquid fuels are so much preferred. The possibility of intimate mixtures of air and gas reduces the necessary amount of excess air as compared with coal. Also the regularity of supply of oil or gas to a furnace permits closer adjustment than would be possible when coal as ordinarily stoked is burned.

While low excess air is generally desirable, the exact amount necessary is quite different in different cases. The vertical lime kiln due to its high temperature in the burning zone which is filled with lime, presents an ideal combustion chamber and can be operated with no excess air whatsoever. Repeatedly has the author obtained zero oxygen and zero carbon monoxide, and very seldom have both been present. There may have been considerable of either present, but the other was always absent. This, in a way, proves that it is not combustion-chamber space that counts but rather effectiveness of combustion space.

In some industries, such as brick making, smoke will deliberately be made; in others, such as enameling, and when semi-muffles are employed, smoke is absolutely fatal to first-class enamel ware. Evidently, if smoke in traces is not permissible, excess air not desirable, and high furnace depreciation objectionable, the job of furnace design then becomes one for a juggler rather than an engineer. Still, by the exercise of proper care a fairly satisfactory arrangement can generally be devised. It is chiefly in such a case that natural gas or oil proves its desirability.

In some cases excess air is objectionable because it reduces efficiency, and high temperature is objectionable because it injures the product. Evidently the only recourse in such a case is retarded combustion. When retarded combustion is to be employed the fuel-burning equipment must be such that there will be only a slow mixture of combustible gases and the air. Then the heat generation and heat absorption take place at the same time, and the temperatures reached are decidedly lower, even with the theoretical amount of air, than they would be if the burners were such as intimately mix the fuel and air. In such cases the desirable size of combustion chamber is one as small as possible and of a kind that will disturb the stratified air-gas streams as little as possible.

In some instances steam is used for atomizing oil or for producing forced draft. Its use is often abused and, due to the excess amount employed and its high specific heat and water content, the furnace gases may be greatly diluted. The argument put forward in such cases is that the use of steam reduces furnace temperatures and so greatly lengthens the life of the refractory. The argument is not always correct, but when it is, unfortunately, the increased refractory life is attained at the expense of fuel economy. In one case, burning oil and with only sufficient excess air to give 3 per cent oxygen in the waste gas—not a bad condition at all—the amount of steam used for atomization was so great that the furnace temperature next to the burner with short-flame conditions was only 2000 deg. fahr. In view of the fact that the working temperature in this case was 1750 deg. and the terminal temperature head 50 deg., it becomes evident that conditions were extremely wasteful, the heat-absorption head being only 200 deg. fahr. and practically all caused by steam, relatively very little by excess air.

It is true that the conventional Orsat is a valuable instrument, but under certain conditions, encountered in industrial-furnace study especially, it should be arranged so that the moisture in the waste gas can be determined also. As it is, this influential factor is often ignored.

Excessive use of steam is also often encountered where gas producers are employed, and many even believe that steam makes better, richer gas. Whatever the effect in the producer may be, it certainly requires fuel to make the steam; and since in such plants the boiler as a rule is considered of secondary importance, it is very wastefully operated. Ordinarily in such plants, of the total coal burned in the producer and boiler, the latter takes about 15 per cent. The theoretical requirement is of course much less, but in practice the author has found 15 per cent common.

Of course, some kind of endothermic agent is necessary to keep down clinkers in the producer, and steam, while expensive, serves the purpose fairly well. But as a rule too much is used. And then, is steam really necessary? Could not CO_2 gas serve the purpose quite as well or better? Where there is a dependable supply of fairly smokeless waste gas from combustion, the mixing of this gas to the extent of 6 per cent CO_2 in the air going to the producer will keep clinkering down better than steam. Such a mixture will

make as much gas, and for many purposes a better gas, due to the much higher carbon monoxide and low hydrogen contents.

The author tried this method and obtained results which were far beyond all expectation. Not only was the boiler coal eliminated, but the kiln capacity and efficiency were increased. The flame characteristics were better and milder, and easier on the product and on the refractories. Before CO_2 was used, the gas from this particular producer contained from 8 to 11 per cent CO_2 and 15 to 20 per cent CO. After steam was replaced by the CO_2 gas, the CO_2 contained in the producer gas was only 6 per cent. The difference is almost fully accounted for by the CO_2 in the air going to the producer, and consequently practically all the carbon in the coal was converted to CO. (When steam is used, some of the CO combines with the oxygen from the dissociation of the steam.)

The method of using CO_2 for this purpose was patented by Eldred some 20 years ago, but the system was not generally adopted, partly because it was not known how to overcome the disadvantages, and partly because the advantages were not realized. It is often objected to because along with the CO_2, nitrogen is added to the air from the waste products of combustion; but this is entirely overbalanced by the fact that steam, of which less weight is necessary, has more than twice as great a heat-carrying, and therefore heat-wasting, capacity as the gas.

APPLICATION OF WASTE-HEAT BOILERS TO INDUSTRIAL FURNACES

Practically all industrial furnaces that are used for processes requiring high temperatures discharge the products of combustion at temperatures sufficiently high for steam-making purposes. If at the same time the hot gases are discharged in large enough quantities, it may be advisable to install waste-heat boilers. A waste-heat boiler would always be justified from the standpoint of thermodynamic efficiency, but in many instances the process is so intermittent, or the investment so great, or the price of fuel for direct-fired boilers so low, that a waste-heat-boiler installation is not justified from a dollar-and-cents standpoint.

The quantity of steam obtainable as a by-product from a given process depends upon the weight or quantity of the gases, their temperature, and their specific heat. In most cases in the United States fuel is so cheap that it is not considered worth while to recover the heat from the gases if their temperature when leaving the process furnace is less than 900 deg. fahr. As fuel becomes higher-priced and heat-reclaiming apparatus more efficient, this minimum temperature will be lowered correspondingly. As long as the process carried on in the furnace does not involve a drying out of the substances, and provided the fuel does not contain large quantities of hydrogen, the gases may be considered dry and their average specific heat taken as 0.25. When hydrogen-bearing fuels are used, this specific heat may be increased 1 or 2 per cent due to the water vapor in the gases. It therefore requires a drop of about 400 deg. fahr. for ten pounds of gas in order to evaporate one pound of water. This immediately indicates the much larger weights of gases it is necessary to pass through the boiler as compared to direct-fired practice. On account of this it is necessary to use induced-draft fans to draw the gases over the boiler surfaces at high velocities so that the quantity of heat transferred per square foot of heating surface will be high enough to operate the boiler at approximately its rated capacity. The proper gas velocity therefore depends upon the heat-transmission characteristics of the boiler. Experience seems to indicate that the straight-tube, horizontal-type, multi-pass boiler of at least three—and preferably four—passes is the one best suited for this work. The high velocities of the gases require uniform cross-sections throughout the boiler to prevent laning and short-circuiting. The bent-tube types of boilers are used in certain cases, but are not nearly as efficient because they lack the characteristics just mentioned.

The question of whether or not superheaters or economizers shall be installed is decided just as in the case of direct-fired boilers. If the steam requirements are such that the steam should be superheated, it is advisable to install the superheaters ahead of the boiler for lower-temperature gases, and within the boiler where the temperatures are higher. If the gases carry considerable dust, as in the case of cement plants, the superheater should be of the bare-tube type on account of the greater ease of cleaning. The value of an economizer installation between the boiler and the induced-draft fan depends upon whether or not the saving of the additional heat will also mean a saving in money. As in the case of large central stations, in some plants economizers would pay, and in others they would not.

A large number of industries have already made use of waste-heat boilers, but only a very few of the more progressive concerns are utilizing the heat from their waste gases. In some cases the operators seem to be very hesitant about installing waste-heat apparatus because of the unsuccessful previous attempts that have come to their notice. It is true that every application is a distinctly individual problem requiring careful study and execution, so that those called upon to do this work should give the matter due consideration in order to insure success. A notable example of this is the oil refinery. About sixteen years ago some Van Dyke boilers were installed in connection with oil stills. The hot waste gases were passed through the Van Dyke water-tube boilers and out through a natural-draft stack. The operators soon found the draft insufficient for their process, so they built another stack at the other end of the oil-still furnace. Perhaps one-third of the gases passed through the boilers and two-thirds out of the new stack. At any rate, within the past year modern waste-heat boilers of the four-pass straight-tube type with induced-draft fans have replaced the Van Dykes. These are delivering a maximum of 950 boiler hp. where the old boiler delivered only 125. The induced-draft fan enables the operators to have much closer control of the process, so that the running time has been reduced by 10 per cent. This means the plant output has been increased 10 per cent without any additional capital expenditure.

Perhaps the cement industry has the greatest number of waste-heat-boiler applications. The cement-burning process is continuous, and therefore the waste-heat steam supply is also continuous. A reasonably tight installation working on the dry process will get from 11 to 12 boiler hp. per barrel of clinker per hour when running at capacity. This may be enough to supply all the power required in the mill, so that direct-fired boilers will be required only when the kilns are not operating. Some wet cement-burning processes are so wet that the gases from the kilns are no higher than 500 deg. fahr. in temperature. This is too low for waste-heat uses, and consequently those mills using this process will be handicapped when the cement industry again becomes competitive after the present era of intensive concrete-road building has passed. Other wet processes give exit gases of from 900 to 1000 deg. fahr., and some of the plants where they are installed are using waste-heat boilers successfully. The temperature of the waste gases from a dry-process kiln may run from 1200 to 1500 deg. fahr., and occasionally with secondary combustion at the kiln mouth the temperatures are considerably higher. The cement-mill operators have found, as in the case of other industries, that waste-heat boilers give better combustion control and increase the output from a given kiln equipment. The boiler units run as high as 1000 rated hp. in the cement-mill installations.

Next in line are perhaps the open-hearth steel furnaces in so far as number of installations are concerned. The boilers, however, are much smaller than in the case of the cement industry, ranging from 300 to 500 rated hp. Since the open-hearth furnace is intermittent in operation, the steam supply is likewise. Therefore the waste-heat application is most advantageous where there are a number of furnaces operated in rotation and the steam supply is consequently fairly continuous. On the other hand, it is an easy matter to arrange the boiler so that it may be direct-fired when the open hearth is down, and this is sometimes done. Many open-hearth operators put in waste-heat boilers to cool the gases so that they can be handled by an induced-draft fan. Unfortunately, a large number of plants are equipped with inefficient boilers merely for the purpose of increasing the open-hearth tonnage and lengthening the intervals between cleaning periods on the checkerwork. In these cases the boiler more than justifies its installation, but with a little more careful design of the heating-surface arrangement it would bring still greater returns, and perhaps make it possible to cut out a direct-fired unit.

There are many other more or less successful installations, such as those in connection with the air furnaces for malleable-iron foundries, billet-heating furnaces in rolling mills and forge shops, enameling ovens, zinc roasters, copper smelters, and zinc smelters; also blue-flame water-gas machines and the larger glass-melting pots.

All of these processes can be arranged so that the waste gases can be utilized for making steam, and more and more plants should be equipped with waste-heat boilers in order to obtain greater economy and efficiency.

INCOMPLETE COMBUSTION

When the working temperature is high the loss due to incomplete combustion becomes greater than at low working temperatures. Assuming pure carbon as fuel and a working temperature of 1600 deg. fahr., no radiation, no dilution due to moisture, no incomplete combustion and only the theoretically required amount of air, then 63.2 per cent of the heat will be available and the remainder will be lost. If all the carbon were burned to CO and there were no other losses, only 12.6 per cent of the heat in the fuel would be available. Under practical operating conditions, when radiation and dilution are such big factors, the heat available becomes 47 per cent in the case of complete combustion and only about 5 per cent in the case of incomplete combustion. In other words, practically all the heat generated in the conversion of C to CO, due to the low temperature rise of the resulting products of combustion, becomes a loss under practical working conditions. This is not so in boiler practice due to the low-temperature working level employed. There the keeping down of oxygen is the most important, while in the industrial-furnace work the importance of carbon monoxide predominates.

Unfortunately, loss due to incomplete combustion is quite common in the case of some furnaces and kilns. In special instances the author has found as high as 15 per cent CO in waste gas from kilns, 5 to 10 per cent being quite common. All the wood-fired kilns are wasting 40 per cent of fuel due to incomplete combustion. While wood-fired kilns are in a minority now, originally all lime was made in them, and this waste has been going on since the time lime began to be used. Wood was burned under lime kilns with a resulting thermal efficiency of about 20 per cent for hundreds of years, but the reason for low efficiency and the remedy therefore were not discovered until only a few months ago. The responsibility rests, of course, upon the owner and on his insistance that dependence be placed upon empiricism rather than upon exact science.

FURNACE DESIGN

Furnace, oven, or kiln design is commonly quite crude. In the past little attention was given to proper gas flow, benefits to be derived from radiation, proper furnace heat capacities, proper heat transfer, convection currents, dead spaces, by-passing and channeling of gases, and other beneficial or harmful factors. In fact, only recently have there appeared any books on the general subject, about the first one being that by Professor Trinks, which was published only in 1923.

Until very recently there was no basis for rating lime kilns. One kiln might have had a capacity twice as great as another, but due to the lack of a capacity standard it could not be known if the increased production was due to the greater size of kiln or to greater production efficiency. It was not until the present year that The National Lime Association formed a committee upon the author's suggestion to create a kiln-capacity standard based upon kiln content.

There are other types of industrial heating apparatus that lack a standard for comparison, and due to this their progress has slowed up a great deal. It is true that in some cases the formulation of standards seems impossible, but an effort at least should be made in all cases.

No matter how the heating capacity of the gas is guarded against reduction by radiation or dilution, full benefit will not be obtained except where sufficient heat-absorbing surface is provided. Full muffles should not be employed except when really necessary. When necessary they should be designed for high heat transfer, and for even heating by providing for proper gas distribution and the relation of muffle surface to furnace required. Heating capacity should be such that terminal temperatures are low—not exceeding 100 deg. fahr. in the case of the high-temperature work.

If at all possible, semi-muffle furnaces are to be preferred; in these the hot gas enters the muffle, but the product which is being heated does not come in direct contact with the main hot-gas stream. With the semi-muffle as well as with the full muffle, heat-radiating surfaces should receive careful consideration. In all industrial-furnace work hot-wall radiation and luminous-flame radiation transfer a very large portion of the total heat.

When fortunately the hot gas can be brought in direct contact with the heating product, the general design should then be such that the gas can flow all around the product at the greatest permissible velocity. None of it should be able to escape without first delivering its portion of heat, and none should be permitted to recirculate and thus reduce the temperature difference and heat transfer. In such instances the furnace should, if possible, be operated under pressure, when the distribution of gas will be more equal, and there will be fewer chances for short-circuiting.

Industrial Furnaces

Comparison of Boiler and Industrial Furnaces—Fuels for Industrial Furnaces—Furnace Efficiency and Its Improvement—Electrically Heated Furnaces—Automatic Temperature Control—Selection of Fuels—Determination of Furnace Sizes, Etc.,

By W. TRINKS,[1] PITTSBURGH, PA.

BY THE TERM "industrial furnaces" we denote structures in which heat is imparted to metallic and ceramic goods; and in which temperatures above 600 deg. fahr. prevail. The term does not include furnaces in which the material is melted. Heating chambers for temperatures below 600 deg. fahr. are commonly called "ovens." The subject of industrial furnaces, as will presently become clear, is big enough to merit a whole book and not a paper. Since, however, such a book is in existence, it will probably be satisfactory at this time to touch on the salient points only and to refer to more extended publications for additional details.

The subject of industrial furnaces is one which has long been neglected by mechanical engineers in favor of boiler furnaces. It is extremely complex on account of the variety of sources of heat energy, and of types of furnaces. As a proof of this statement the tabulation at the top of the following page, which was originated by the W. S. Rockwell Co., furnace engineers of New York City, is offered.

The present paper differs from the majority of papers offered to the Society in that it is not limited to reporting recent progress in a subject with which every mechanical engineer is familiar. On the contrary, it is intended to be educational and is an attempt to interest the mechanical engineer in a live subject which up to the present time has not been claimed specifically by any branch of engineers, with the possible exception of the metallurgists. While the latter are essentially interested in the product of industrial furnaces, the engineering of this equipment may be claimed rightfully by the mechanical engineer, because the heating of steel or other metals without any chemical change or without melting is part of what is commonly known as "mechanical technology."

Comparison of Boiler Furnaces and Industrial Furnaces

Since mechanical engineers have given so much attention to the boiler furnace, it may be advisable to introduce the subject of industrial furnaces by a comparison of the two types. In boiler furnaces we produce heat by that fuel which will make steam at the lowest possible cost. The latter involves B.t.u. developed per dollar's worth of fuel, interest on investment, labor of fuel preparation, and labor of fuel feeding and ash removal. In the industrial furnace we wish to produce "perfectly heated" material at the lowest possible cost. In this requirement a new factor enters, namely, the condition of the heated material. If the material is not heated uniformly, differences in hardness, strength, and ductility occur, which spoil it for its final use. If the material is burnt or scaled, it may also become worthless for final use. In consequence, the factor which enters into industrial furnaces and which does not exist in boiler furnaces is "quality of heated product." In boiler furnaces sulphur does comparatively little harm, whereas the use of fuels containing appreciable quantities of sulphur is seldom permissible in industrial furnaces. Furthermore excess air is permissible in boiler furnaces and is frequently necessary, whereas it is utterly inadmissible in most industrial furnaces on account of the scaling, or oxidation, of the charge. On the contrary, reducing atmospheres are usually wanted, that is to say, atmospheres which contain unconsumed combustible.

A further difference between the two types of furnaces lies in the pressure relation. In boiler furnaces a slight vacuum or a draft is permissible and often exists. In industrial furnaces a draft (i.e., a partial vacuum) is out of the question, because it would cause an inflow of air and consequent oxidation of the charge. The pressure immediately above the hearth of the furnace must always be slightly greater than atmospheric, the excess pressure usually amounting to from 0.02 in. to 0.025 in. of water. One result of this condition is a heat distribution in the brick walls decidedly different from that in the walls of most boiler furnaces. Not only the joints of the walls but the bricks themselves are porous, and in consequence in the walls of industrial furnaces there is an outward flow of heated gases in addition to an outward flow of heat by conduction; whereas in most boiler furnaces, while heat flows outward, cold air flows in to a certain extent through the walls. The difference in brick temperatures thus produced is shown in Fig. 1.

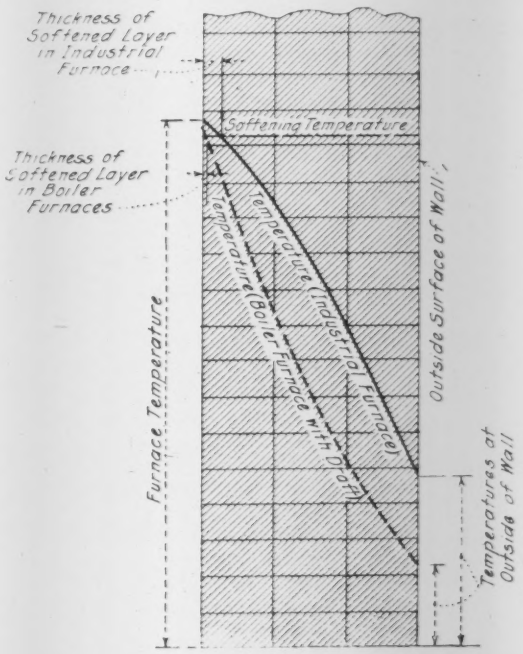
Thickness of Softened Layer in Industrial Furnace
Softening Temperature
Thickness of Softened Layer in Boiler Furnaces
Temperature (Industrial Furnace)
Temperature (Boiler Furnace with Draft)
Outside Surface of Wall
Furnace Temperature
Temperatures of Outside of Wall

Fig. 1 Diagram of Temperature Distribution in Brick Walls of Industrial Furnaces and Boiler Furnaces

Fuel for Industrial Furnaces

On account of the close interaction between the nature of the fuel and the quality of the heated product, the study of fuel for industrial furnaces is a vast and extended subject. Among the factors which must be considered in the selection of a fuel or other source of heat energy in an industrial furnace, the following are of importance. First, constancy of source of heat. If the fuel varies in composition, it is almost impossible to maintain a constant temperature and constant atmosphere in the furnace. Second, if the fuel is of such a nature that it clogs up the burners, the maintaining of a constant furnace temperature and constant furnace atmosphere is extremely difficult if not impossible. Besides, a great deal of attendance is needed in that case to maintain regular operation. Third, if the fuel burns with an exceedingly bright and luminous flame, overheating of parts of the charge is very likely to happen, unless special precautions in design and operation are taken. In furnaces carrying temperatures of 1800 deg. fahr. or less, such fuels can as a rule be used only by wasting a considerable part of the heat. Fourth, the fuel should be such that it can burn in a reducing atmosphere without excessive smoke. Fifth, it is desirable to use a fuel which can be distributed to a number of scattered furnaces without excessive cost. Sixth, the fuel should be of such a nature that there is little chance of interruption of supply. In changing over from one fuel to another, there is usually much spoil-

[1] Professor of Mechanical Engineering, Carnegie Institute of Technology. Mem. A.S.M.E.

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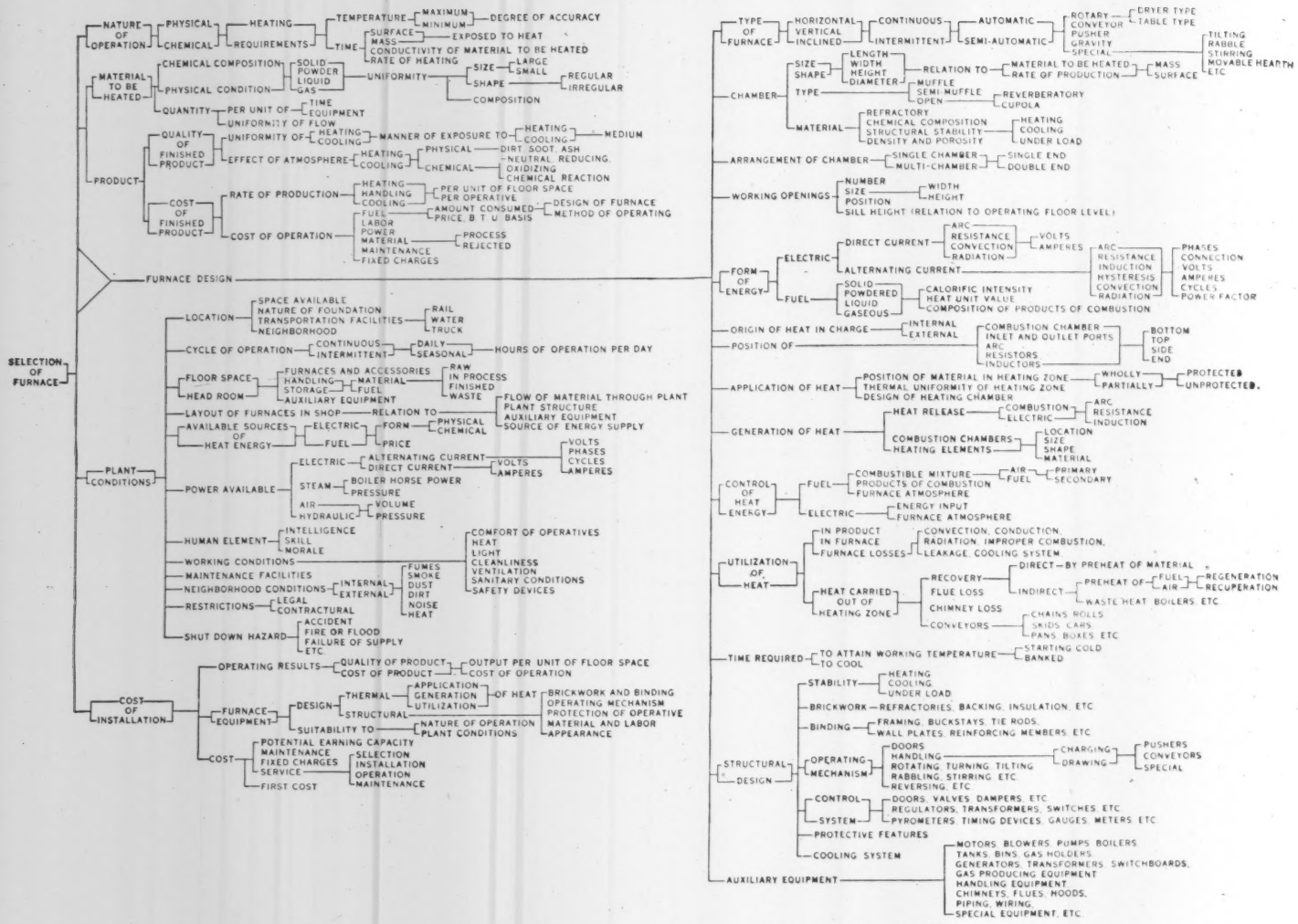

FACTORS GOVERNING SELECTION OF FURNACES FOR INDUSTRIAL HEATING OPERATIONS
(Reproduced by courtesy of W. S. Rockwell Co., New York, N. Y.)

ing of heated material, and the cost of rejections runs high. Seventh, fuels should not cause any deposits of ash, carbon, or other foreign matter on the heated material. Eighth, the fuel should be such that no excessive stand-by losses are caused during periods of shut-down.

Temperature Control—Furnace Efficiency

In late years great strides have been made in the science of heat treating, and metallurgists have realized that very close temperature control for quenching is essential in the production of reliable heat-treated parts. In this connection it should be mentioned that temperature control is very difficult with a fuel of varying quality. This is the reason why furnaces fired with city gas and furnaces heated by electrical energy have increased in numbers so rapidly within the last few years. It should be noted that city gas and electrical energy involve the greatest cost on the B.t.u. basis. The only conclusion to be drawn from this fact is that in the United States the cost of rejections and the cost of labor of attendance are of vastly greater importance than the cost of the fuel. If we denote by thermal efficiency the ratio of heat in the product to the potential heat in the fuel burned to heat the product, we find another very great difference between boiler furnaces and industrial furnaces. The thermal efficiency of boiler furnaces is quite high and ranges in practical operations between 60 and about 90 per cent. The thermal efficiency of industrial furnaces is usually quite low, and lies between 10 and 30 per cent in the general run, with lower and higher values existing every now and then. The highest recorded efficiency is about 55 per cent to 60 per cent, while many furnaces operate at not over 5 per cent efficiency. The latter varies not only with the kind of heating but also with the design of the furnace. Of two furnaces of different design, both doing the same kind of work, one may easily use three times as much fuel as the other for heating a given charge. The efficiency varies also with the rate of driving the furnace, as shown in Fig. 2, which applies to a continuous furnace.

One reason for the great difference in thermal efficiency is immediately apparent. It lies in the low temperature of the material to be heated in the boiler furnace and the high temperature of the charge in the industrial furnace. (The temperature ranges for various heating operations in the treating of steel are marked in Fig. 3.) Gases can give up heat only as long as they are hotter themselves than the material to be heated. In consequence, the flue gases must leave industrial furnaces at a very high temperature. On the basis that the products of combustion leave an industrial furnace at a temperature of 50 deg. fahr. in excess of that of the charge, two curves have been shown in Fig. 3, giving the maximum possible thermal efficiency of an industrial furnace.[1]

The curves of Fig. 3 do not apply to continuous furnaces. These curves are plotted against temperature of charge. One curve is shown for clean producer gas, whereas the other is shown for oil. The curves were calculated on the basis of simple furnaces and of no preheat in the fuel or in the air. The actual thermal efficiency of industrial furnaces is as a rule much lower than the values given in Fig. 3. The discrepancy is particularly noticeable in low-temperature furnaces. The reason for the great difference between the highest possible thermal efficiency and the actually lower thermal efficiency of the low-temperature industrial furnaces lies in the fact that the products of combustion must be cooled down before they

[1] It is true that in intermittent furnaces which heat up with the charge, during part of the heating time the gases pass out at a temperature lower than the final temperature of the charge; and if only this feature were to be considered, an ideal efficiency *higher* than those shown in Fig. 3 would be possible. But in such furnaces the loss caused by dissipation of heat stored in the brickwork by cooling before a fresh charge is put in, always much more than offsets the gain by extraction of somewhat more heat from the gases.

are allowed to come in contact with the charge. Otherwise the latter will be seriously overheated and burnt. In this respect the industrial furnace differs from the boiler furnace. In many furnaces a great deal of heat is dissipated during the process of cooling down the products of combustion. The amount of useless dissipation varies with the design of the furnace. In consequence of this variation, the thermal efficiency of furnaces varies a great deal. The two lower curves of Fig. 3 include a band of probable efficiency of industrial furnaces in which there is no salvage of heat.

An additional reason for the low efficiencies of industrial furnaces lies in the amount of potential heat which is still contained in the leaving products of combustion on account of the large amount of combustible matter which is usually maintained in these products for the purpose of protecting the stock or charge against oxidation. In the heating of steel, for instance, a CO content of from 2 to 4 per cent is frequently considered necessary for protection, with the

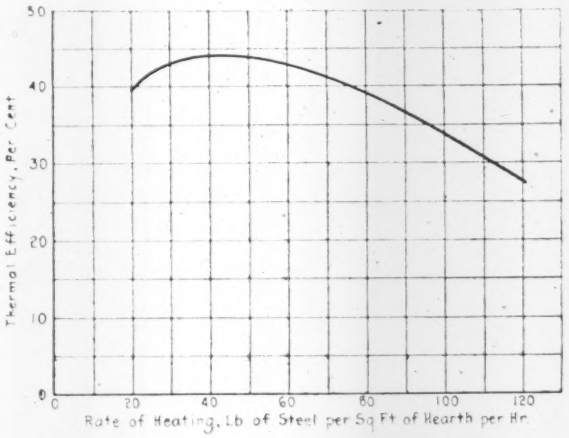

FIG. 2 THERMAL EFFICIENCY OF A CONTINUOUS FURNACE HEATING MILD STEEL TO 2200 DEG. FAHR.

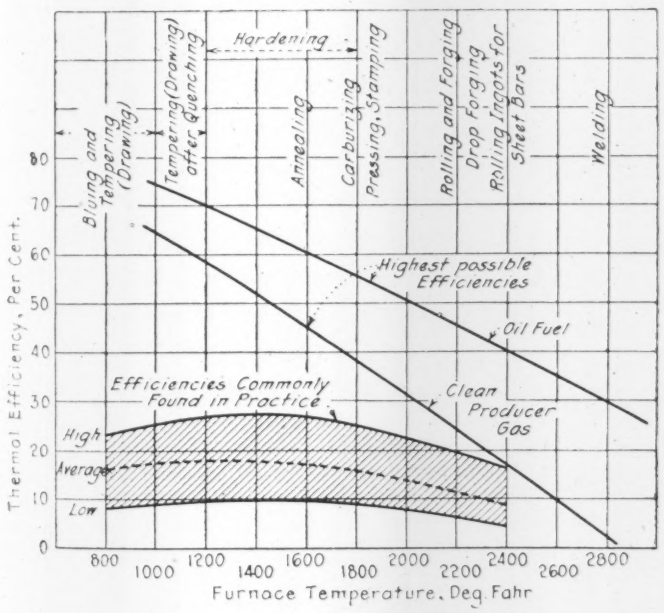

FIG. 3 RANGE OF EFFICIENCIES OF INDUSTRIAL FURNACES OF THE INTERMITTENT OR BATCH TYPE

result that from 5 to 12 per cent of the heat of the fuel is lost by the combustible matter in the waste products.

IMPROVING THE THERMAL EFFICIENCY OF INDUSTRIAL FURNACES

It is obvious that this enormous waste of fuel called for efforts on the part of designers and inventors to improve the thermal efficiency of industrial furnaces. Several ways are in existence for this purpose. They consist in preheating the stock or charge, in preheating the air which is needed for combustion or else the fuel, or both, and finally in the utilization of the waste heat for the generation of steam in so-called "waste-heat boilers." A study of the various systems of heat salvage by preheating the stock in continuous furnaces, by preheating air and fuel in regenerative and recuperative furnaces, and of waste-heat boilers is extremely interesting but exceeds the limit of this paper. Besides, information on this subject is contained in many publications. It is, however, appropriate to state that the problem of when to use either or any of these heat-salvaging devices and which detail construction is suitable for the case in hand, is a subject for long and deep study, particularly on large furnaces with high fuel consumption. It

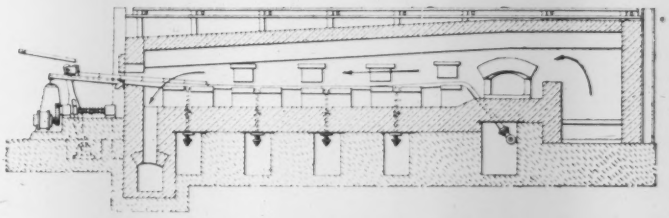

FIG. 4 CONTINUOUS TYPE OF FURNACE

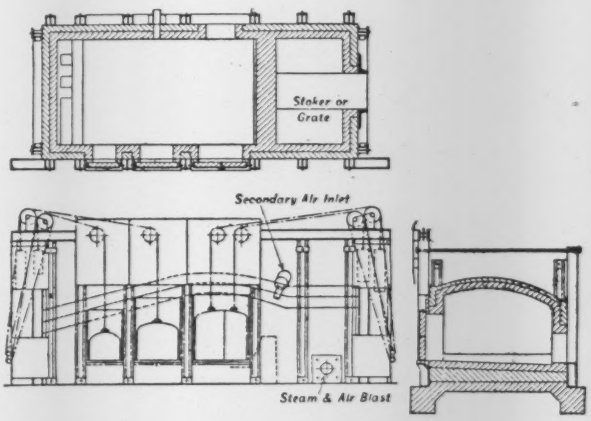

FIG. 5 INTERMITTENT TYPE OF FURNACE

may furthermore be remarked that heat salvage by regeneration or recuperation is not an unmixed blessing. The tending to reversing valves, burning out or cracking of recuperators, and uneven temperature distribution in furnaces due to rapid combustion in highly preheated air, are features which deter engineers from the liberal use of heat-salvaging equipment.

It may be well to point out that the type of furnace the operation of which is most nearly analogous to that of the boiler is the continuous furnace, in which the stock to be heated is charged at one end and leaves at the other as in Fig. 4, flowing through the furnace continually. The gases flow in the opposite direction to the stock, giving up heat as they go; hence their final temperature is considerably lower than the final temperature of the stock.

After the continuous furnace is started, it operates steadily with little change of temperature. In furnaces of the intermittent type, such as Fig. 5, on the contrary, especially those for heating pieces of large size, it is often found necessary to allow the furnace interior to cool down after the heated pieces have been withdrawn, before inserting a cold charge, because otherwise excessive temperature differences would be produced in the heated stock, and would result in cracks and other defects.

This brings out another difference between the operation of the boiler furnace and that of industrial furnaces. In boilers the tubes can usually absorb heat from the fire as fast as it can be produced, and excessive rates of heat development do no harm. In industrial furnaces, however, the rate at which heat can be developed is often sadly limited by the rate at which the heating stock can absorb it. A particular example of this occurs in heating large steel ingots for forging. Steel passes through a "blue brittle" range at about 500 to 700 deg. fahr., and if large ingots are heated too quickly through this range, excessive temperature differences are produced in the interior of the ingot, resulting in "clinks" or cracks, which may later cause failure of the forging produced from it. Cracks do not occur in small pieces or in pieces of very soft steel, but if these are heated

at an excessive rate, as is sometimes done for drop forging, the surface may actually be melted while the interior still has a "bone" or relatively cold core.

Electrical Energy as a Source of Heat in Industrial Furnaces

In the last few years the use of electrical energy as a source of heat in industrial furnaces has made enormous strides. At first thought it may appear extremely wasteful to use so expensive a form of energy as a heating medium in industrial furnaces, but upon reflection it will be seen that it has many advantages. First of all, no heat is taken out of the furnace by means of the products of combustion. This feature insures to the electrically heated furnace a greater efficiency from electrical energy to heat in the charge. Second, power plants are becoming more and more efficient, so that a greater percentage of the heat in the fuel is present in the electrical energy transmitted. Third, no products of combustion are discharged into the shop. Fourth, the furnace can easily be placed in the production line. Fifth, temperature control within quite narrow limits is easily attainable without any attendance whatsoever. For many processes such as heat treating, drawing and tempering, bluing, japanning, etc., the electrically heated furnace has proved to be a very serious competitor of the combustion-type furnaces. Electrically heated furnaces are shown in Figs. 6 and 7.

Although thermal efficiency and fuel economy are of importance for large furnaces, they lose much of their importance in connection with the many industrial furnaces which serve in the mass production of small and medium-sized pieces. In furnaces for this class of work, as a result of the peculiar situation of the United States with regard to cheapness of fuel and dearness of labor, the cost of fuel in the industrial furnace is so small a fraction of the total cost of the finished product that it has practically no influence upon the sales price. Cost of labor in tending the furnace and cost of spoiled and imperfect material are of vastly greater importance. Definite figures on these costs, however, have not been so accessible to theorists as fuel costs nor so amenable to calculation, which is perhaps one of the reasons why fuel economy has been stressed almost exclusively. Actual cost records compiled over considerable periods of time show that the fuel cost of heat treating is frequently less than one per cent of the manufacturing cost of the finished article. For very light parts it is less than one-fifth of one per cent.

As a result of this situation it will be found that many manufacturing processes are carried out by the use of heating machines, that is, industrial furnaces in which the material-handling and conveying equipment by far overshadows the heat engineering. A few examples of such machines are shown in Figs. 8, 9, and 10.[1] Machines of this type are used, for instance, in the heat treating of steel balls, file blades, razor blades, pliers, chisels, tools of all sorts, in the heating of parts to which other parts are to be welded, and for many industrial operations. In the design of the labor-saving features of the more complicated heating machines mechanical engineering is involved to a much greater extent than heat engineering. An idea of the mechanism required may be obtained from the illustrations named, as well as from Figs. 11 and 12, which show the charging mechanisms of such furnaces.

Automatic Temperature Control

On account of the high cost of spoiled, burnt, or irregularly heated material, American shop managers and superintendents recognized at an early date the wisdom of automatic control of furnace temperature and furnace atmosphere. While automatic control of furnace temperature is generally considered to be a modern invention, brought about by the necessity of close temperature control in making gun forgings and other war material during the Great War, it should be noted that a very successful automatic temperature control for gas-fired furnaces was put on the market in the United States as early as 1908 and that the same control is being sold today in large numbers without any change whatsoever. The general principle of automatic temperature control is extremely simple. A thermometer or pyrometer, by means of one or several relays, controls the supply of heat energy. In the concrete embodiment of this principle, however, many difficulties occur. Location and protection of the pyrometer have to be considered; the control of heat energy without "hunting" required much study and experience; and, last but not least, it required time to realize the fact that control of *furnace* temperature does not necessarily result in the desired control of the temperature of all parts of the *charge*. The upshot is that while a great deal of engineering has been done in the design and in the construction of equipment for automatic control of furnace temperature, even more careful engineering must be done right along in the application of such equipment if good results and properly heated material are to be obtained. A temperature-control device for furnaces is shown in Fig. 13.

General appreciation of the fact that in the heating of many materials control of furnace atmosphere—that is to say, control of the composition of the gases in the furnace—is just as important as temperature control, is of more recent date. Many of the devices for automatically controlling furnace temperature also control furnace atmosphere, if the composition of the fuel and the pressure at which it, as well as blast air, is delivered remain absolutely constant. This statement holds true with regard to gaseous fuels. In liquid fuels another factor enters, namely, the temperature of the fuel, which controls the viscosity. In addition, clogging of the fine opening through which the liquid fuel is atomized frequently alters both temperature control and atmosphere control. If solid fuel is burned on the grate or in pulverized form, the control of temperature and of atmosphere becomes still more difficult. Whereas one and the same principle is the basis of the various systems of temperature control, the atmosphere-control devices utilize several different principles, among which are: the balancing against each other of two pressure differences, one dependent on the rate of fuel feed and the other on the rate of air supply; the entrainment of fuel by the moving stream of air, or vice versa; the proportional metering of fuel and air; and control of air flow by the flue-gas analysis.

In the application of the first three principles the success of atmosphere control depends on the constancy of pressure, temperature, and composition of the fuel. It is least difficult in the case of gaseous fuels, more so for liquid fuels, and quite difficult for solid fuels. A complete description of the different temperature-control devices available and of their application would far exceed the limits of this paper. One example of the many types is shown in Fig. 14.

Selection of Fuels

Plant engineers are frequently called upon to decide as to the type of furnace to be installed for a given heating operation, and the kind of fuel or heat energy to be used. Within the limits of this paper it would be impossible to make an exhaustive study of different fuels and furnace types, but certain pointers can be given which may be of some assistance in making selections.

In general, it is required that the furnace should heat a given weight of stock in a given time to a given temperature; should heat it in such a manner as to give the desired properties in the product; and should do it at the least cost per unit of finished product. The last requirement includes many separate items such as cost of a given quantity of heat in the fuel (which, again, may vary greatly during the life of the furnace); the cost of delivery of fuel to the furnace and removal of waste products, such as ashes; wages to heaters and cost of supervision of furnace; availability of fuel and cost of changing over from one fuel to another; cost of shutdowns due to lack of orders (payroll and service charge to be carried during shutdown); cost of charging and emptying furnace, of labor in handling materials, and of heat losses of furnace; cost of furnace maintenance; cost of rejections or of reduction in grade of finished material; cost of scaling and burning; costs of processes which remove defects originating in the furnace; cost of delays due to interruption of fuel supply, to adjustment of burners, or to furnace troubles; cost of equipment which is necessary to insure regular fuel supply; cost of reduction in output due to improper location of furnace in factory and due to the effect of excessive heat radiation or obnoxious fumes on the workmen; fixed charges for furnace investment, including that for auxiliaries, foundation, and stack; and adaptability of furnace to changing conditions. It is usually impossible to predict beforehand with any accuracy the cost of all of these different items. The best that can be done is to estimate

[1] Figs. 10, 11, and 12 are published by courtesy of the American Gas Furnace Co., Elizabeth, N. J.

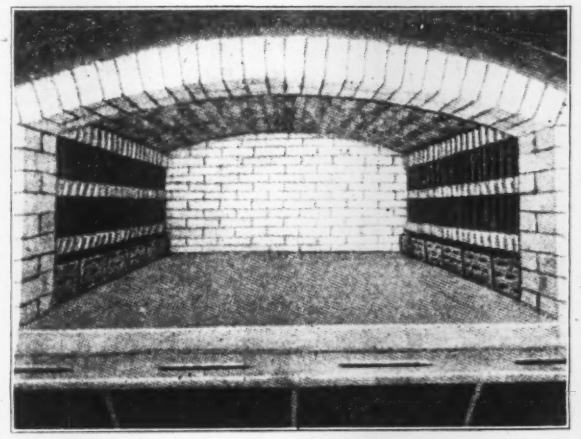

FIG. 6 ELECTRICALLY HEATED FURNACE WITH RIBBON-SHAPED HEATING ELEMENTS ON THE SIDE WALLS

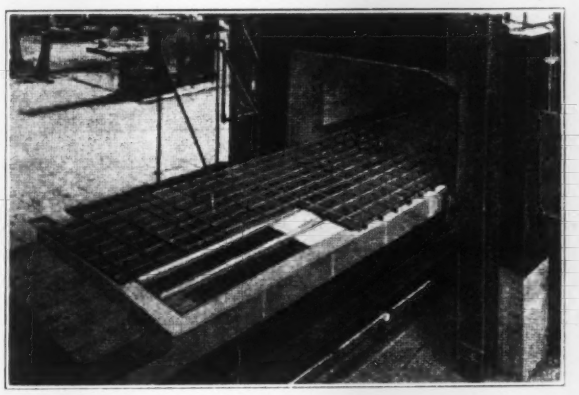

FIG. 7 ELECTRICALLY HEATED FURNACE OF THE CAR TYPE

(Heating elements are behind muffle plates in the side walls and under hearth plates in the car.)

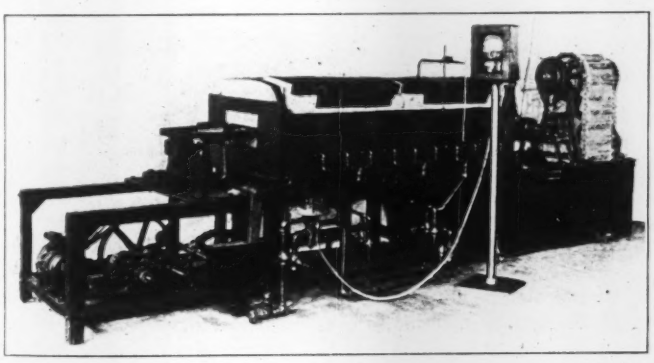

FIG. 10 HEATING MACHINE OF THE RECIPROCATING-HEARTH TYPE USED FOR HARDENING NEEDLES, PENS, SMALL TOOLS, SCREWS, NUTS, SPRINGS, ETC.

(Note the mechanism for driving, and for reciprocating the hearth, also the automatic temperature controller and the conveyor for removing parts from the quenching bath.)

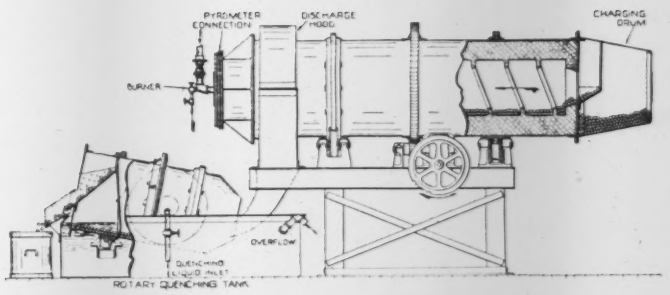

FIG. 8 AUTOMATIC HEATING AND QUENCHING WITH ROTARY FURNACE AND QUENCHING TANK

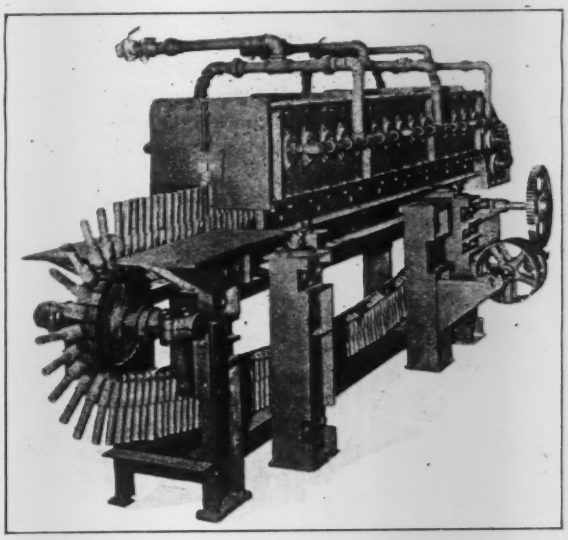

FIG. 9 AUTOMATIC FURNACE FOR HEATING PIECES AT ONE END ONLY

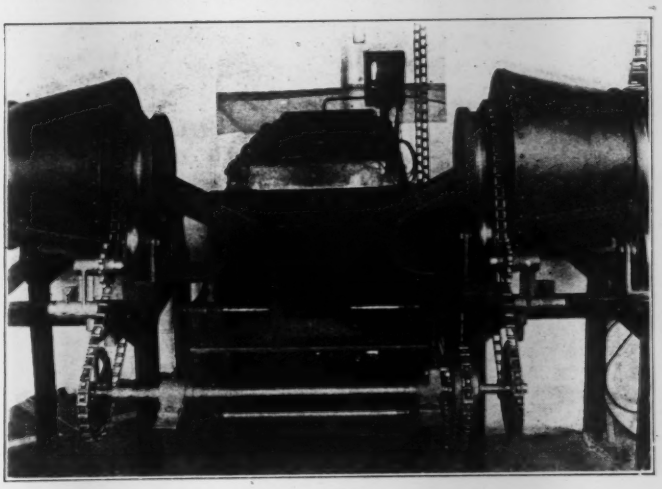

FIG. 11 AUTOMATIC FEEDING DEVICE FOR THE FURNACE SHOWN IN FIG. 10

the relative importance of the chief ones for the particular heating operation which is to be performed, and with these factors in mind to depend on judgment in making the selection.

The factors to be considered in the selection of the fuel were mentioned in the early part of the paper. Their relative importance depends largely on the kind of heating operation and its requirements. For instance, in the heat treating of many small parts, where fuel cost is a very small fraction of the total cost and is far overbalanced by the necessity for absolute uniformity of products and for scale-free surfaces, the energy sources considered are electrical energy, city gas, natural gas where it is available, and water gas or clean producer gas in very large plants. At the other end of the scale of the fuels is coal, which is burnt either on the grate or in powdered form and which, in general, is used for rough work. It is very seldom applied for the finer classes of work. One exception is that of the sheet furnace for heating sheets or tin plates; for this purpose powdered coal has been used ever since the beginning of the industry, although it can hardly be said to be ideal for the work. It is especially suitable for the heating of heavy work such as forging ingots or blooms, in which quality of the surface and deposits of ash on the work do not matter.

Oil fuel is intermediate between (on the one hand) electricity and clean gases of constant quality, which are suitable for fine work, and (on the other hand) fuels such as powdered coal or gases of variable quality, which are suitable only for rough work. Excellent heating of fine work can of course be done with oil, but the result depends to a considerably greater extent on the watchfulness of the operator and on the design of the furnace and proper application to it of suitable burners than in the case of gaseous fuels. Usually the cost of oil also is intermediate between that of the

high-class fuels and the coal, but the cost depends, of course, upon the location of the plant. Although oil appears to be losing ground for the heat treating of small pieces, it continues to be used and will be used to a great extent for larger pieces, such as shafts, disks, etc. Where the mass of material to be heated is large, the fuel cost is a greater proportion of the total and scale on the surface is much less detrimental than in the case of small pieces.

Natural gas occurs in so few localities that the use of this otherwise ideal fuel need not be considered here.

Fuel Requirements for Special Cases

In a general paper like the present one it may appear out of place to mention fuel requirements for special cases. Nevertheless, one or two examples will be given in order to show the care which must be exercised. In heating sheets, for example, it has been found that a certain proportion of water vapor is required in the atmosphere of the furnace in order to preserve the desired kind of thin, uniform scale which will prevent the sheets from sticking during rolling. For such use only fuels containing hydrogen or hydrocarbons would be suitable. Bituminous coal on the grate is still used to a very great extent for this purpose, even though it requires a very careful and skilled heater for producing the desired results. The use of coal in a deep fuel bed is partly due to the necessity for a very uniform furnace temperature only slightly in excess of that of the heated sheets. As a second example the case may be mentioned in which a very high temperature must be produced locally while heat is flowing away from the hot spot to colder parts of the same piece. A typical instance is a furnace for welding the end of one tube to another. On account of the rapid dissipation of heat there must be in this case a very high temperature potential between the products of combustion and the material being heated. Water gas or oil are time-honored fuels for this purpose. It may be noted that fuels *have been* used for purposes for which they are generally considered unsuitable. The results, however, have in general been unsatisfactory, and the mere fact that a given fuel has been used for a given purpose cannot be taken as a guarantee that it is satisfactory for that purpose.

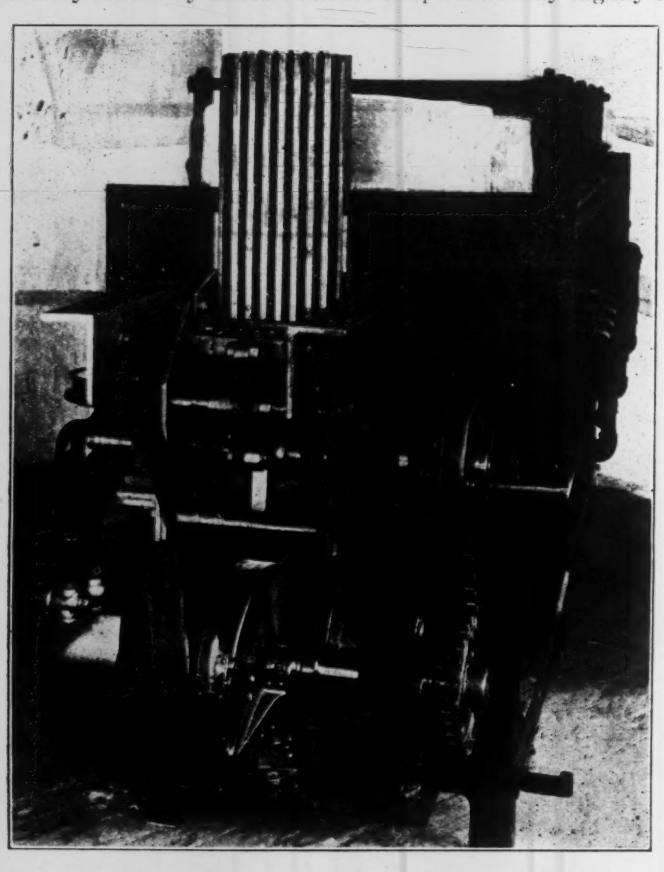

Fig. 12 Alternative Type of Charging Mechanism for Pusher-Type Furnace Used for Hardening Helical Springs

The availability of the fuel also has an influence on its selection for a given use. In steel plants which also contain blast furnaces, fuels which are available at practically no expense are coke-oven gas, blast-furnace gas, and coal tar. These fuels are therefore used for almost all heating purposes around the steel plant, in order that outside fuel may not have to be purchased.

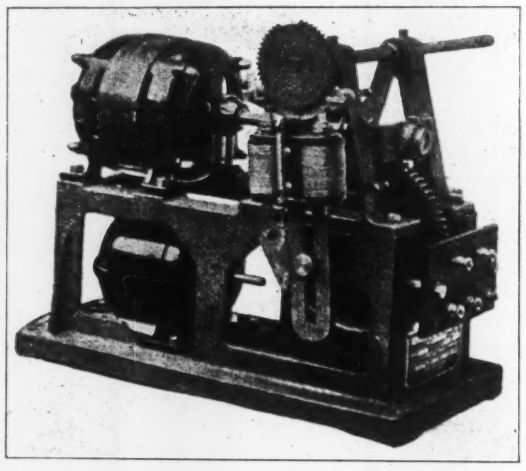

Fig. 13 Example of Apparatus Used in Automatically Controlling the Temperature of Fuel-Fired Furnaces

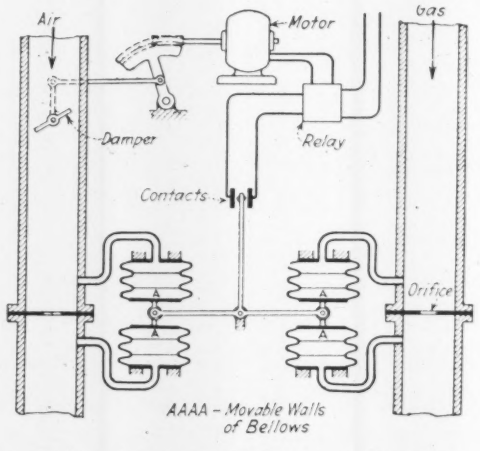

Fig. 14 Example of Apparatus Used in the Control of Furnace Atmosphere

(The pressure drops through two orifices are balanced against each other. Disturbance of the ratio of flow causes a damper in the air line to be moved.)

It should also be noted that the question of suitable labor enters here, inasmuch as in order to obtain satisfactory operation with a fuel which is not particularly suited for the given kind of heating, labor of a higher quality is required than with a more suitable fuel.

Selection of Type of Furnace

While thought must be given to the selection of the fuel or source of energy, the proper selection of the type of furnace requires even more care. To a certain extent the type to be selected depends upon the kind of heating operation to be performed. Often this is not definitely known beforehand. The furnace salesman and the plant engineers agree upon a rough estimate as to what is to be heated in the furnace, but the latter is sometimes used for an entirely different purpose later on. It is not to be wondered at that furnaces installed in this way are found to be unsatisfactory. With uncertainties like the one just mentioned in existence, and because of the restrictions imposed by local plant conditions, it is obvious that only a few pointers can be given here as to the suitability of a given type of furnace.

Where very uniform heating or uniform distribution of temperature in the charge is required, electrically heated furnaces with heating elements practically surrounding the charge are frequently employed. If the furnaces are of the combustion type, the underfired (Fig. 15) or the muffle (Fig. 16) furnace is usually chosen.

Both of these furnaces heat more uniformly than direct-fired furnaces (Fig. 17), but they sacrifice capacity in order to obtain uniformity of heating. Further, they offer difficulties with regard to strength and durability if temperatures in excess of 1600 deg. fahr. are to be maintained. The muffle is also used where it is necessary to control the atmosphere coming in contact with the charge and to prevent the combustion gases from affecting it. The direct-fired furnace in which the combustible is fired directly into the heating chamber and the gases, either completely or only partly burnt, circulate directly around the charge, is most suitable for heating to comparatively high temperatures as, for instance, for rolling, forging, or welding, where oxidation of the surface and uneven temperature distribution do comparatively little harm. It may be noted that at high temperatures radiation is very active and does not permit great differences of temperature to exist except in cavities which are inaccessible to radiation. The side-fired furnace (Fig. 18) is in reality a variation of the direct-fired furnace, but usually has a bridge-wall which serves the purpose of deflecting the gases to some extent. With oil fuel, it protects the charge from the oxidizing and burning effect of the oil flame. The heating is likely to be somewhat less even in a low side-fired type than in a tall direct-fired furnace, but in the side-fired type there is a possibility of using recirculation of the furnace gases, which helps to equalize the temperature. Side firing is used instead of under firing if the load to be carried per square foot of hearth is very heavy.

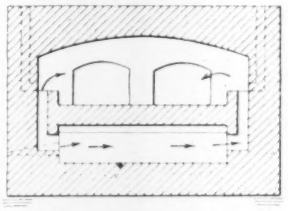

FIG. 15 UNDER-FIRED FURNACE

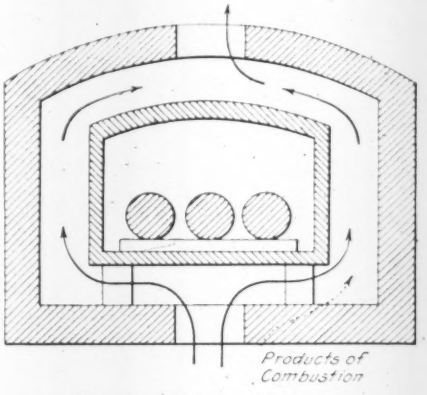

FIG. 16 MUFFLE FURNACE

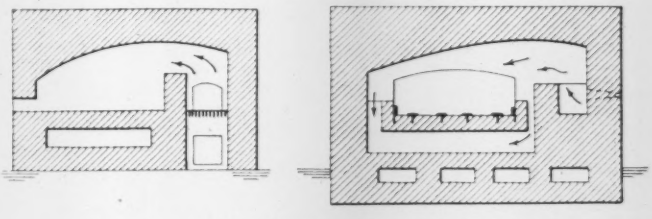

FIG. 18 COAL-FIRED AND SIDE-FIRED FURNACES

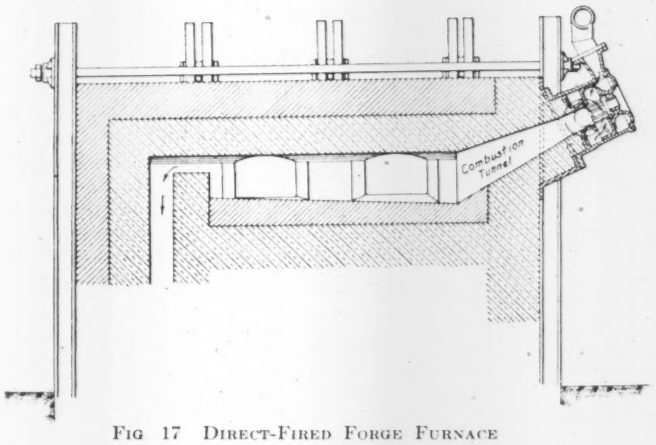

FIG 17 DIRECT-FIRED FORGE FURNACE

The over-fired furnace (Fig. 19) has a perforated arch above which the flame is developed. This furnace is used only for oil firing, its purpose being to protect the charge from spotty radiation of bright oil flames and to reduce the temperature gradient from the high-temperature oil flame to the charge, and thus produce even heating. It has, however, the defect that cold spots are likely to occur on the hearth unless the charge is raised above it on supports.

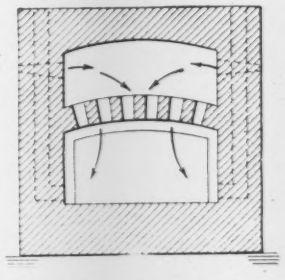

FIG. 19 OVER-FIRED FURNACE

The continuous furnace is very desirable for use in the heating of large quantities of pieces which are exactly or very nearly similar in shape and size. It is not suitable for intermittent operation, for heating pieces of different kinds and sizes, or for heating very thick pieces, because in pieces over about 8 in. in thickness the temperature differences are likely to be too great for satisfactory forging or rolling unless a long hearth is provided on which the pieces are turned over for "soaking"—that is, temperature equalization—or unless another soaking furnace is provided into which the pieces taken from the continuous furnace are charged in order to allow the temperatures to equalize in them.

TABLE 1 HEATING RATES FOR STEEL IN FURNACES OF DIFFERENT TYPES

Kind of furnace	Usual heating rate for steel, lb. per sq. ft. per hr.
Forging (in-and-out type)	80
Heat-treating	30
Annealing	30-60
Under-fired	30
Muffle	15
Electric	45

Lead and salt baths have the advantage of heating steel up to quenching and hardening temperatures with perfect freedom from scaling. In the heating of complicated pieces in such baths, cracking can be prevented by oven preheating to about 800 or 900 deg. fahr. Cyanide baths, of course, are used for hardening.

CALCULATION OF FURNACE SIZES

As a guide in determining approximately the furnace size, figures are given in Table 1 for the heating rates which are usual in heating steel in furnaces of different types. The heating rate is expressed in pounds of material heated per square foot of hearth area per hour. Copper and brass can be heated at a higher rate. For continuous furnaces the most economical rate for heating steel is about 40 lb. per sq. ft. per hour; the usual rate in operation in most cases is about 60 to 80 lb., but in some cases where the furnaces are driven very hard, a rate of 125 lb. is reached. The latter, however, means excessive fuel consumption and excessive cost of furnace repairs.

The present paper has already assumed an almost undue length and yet important considerations affecting industrial furnaces have scarcely been touched upon, such as, for instance, the heat-liberation rate per cubic foot for different fuels and different combustion conditions, heat transfer from the products of combustion or heating elements to the charge and temperature equalization in the interior of the charge. Several papers could be devoted to the theory of recuperation and regeneration, and to the strength and durability of furnace parts. The science of temperature control and of control of furnace atmosphere has progressed to such an extent that a paper could well be devoted to this part of the art alone. Much ingenuity has been expended in the design of equipment for moving the material to be heated through furnaces with a minimum of labor.

As previously stated, it is out of the question to treat all of these features of industrial furnaces adequately in one paper. Reference must therefore, be had to other publications on the subject.

Recent Advances in Methods of Glue Evaluation

Difficulties Presented by Problem of Glue Testing—Viscosity and Jelly-Strength Tests, and Standard Methods for Making Them

By WILBUR L. JONES,[1] MADISON, WIS.

THE subject of glue has always been shrouded in mystery. Erroneous ideas about its origin, properties, and uses have been prevalent from the very beginning; and as for the nature of the phenomenon of adhesion, that has been considered so profound a mystery that men have refused even to speculate about it. Woodworkers have used glue ever since the building of the pyramids, but they never have had the slightest idea as to why it served their purpose. No one would venture to say what the characteristics of a glue were that made it useful for joinery. Not knowing what to require of a glue, buyers were unable to test it or specify it intelligently.

The author believes that the case of glue is quite unique in that the properties that give it utility have never been clearly recognized. Every one knows that steel must have a certain tensile strength, so means were found for determining it. Coal must have a certain heating value, so calorimeters were adapted to the needs of coal analysis. Linseed oil must have the property of being able to dry to an elastic film, so chemists found a way of determining the degree of its unsaturation. But no one knew exactly what glue should have or be in order to give a good joint. Consequently the development of testing methods has been exceptionally slow and groping.

Difficulties Presented by Problem of Glue Testing

The problem of glue testing presents most serious difficulties. In the first place, chemical analysis tells almost nothing. Glue is not a distinct chemical individual; it is a mixture of decomposition products. The quality of a glue does not depend on the presence or absence of any particular compound. The only thing that can be said about it with assurance is that it is generally considered a mark of inferiority if a glue contains a large amount of those compounds that indicate that the decomposition—strictly speaking, the hydrolysis—of the parent substance, collagen or ossein, as the case may be, has been deep-going and complete. Some day it may be possible to grade glue chemically, according to the stage to which the hydrolysis has progressed; but at present we do not know enough about the technical properties of the various decomposition products of collagen and ossein to set up chemical criteria.

The greatest interest has always been centered in the department of physical testing. This is perfectly justifiable, since it is the physical properties of glue that give it industrial importance. We are just beginning to learn something about why glue sticks. The function of glue in joinery, according to our best understanding of it, is describable in physical terms.[2] We know, for example, that it must be of such a consistency as to be able to enter the pores and ducts of the wood. We know, also, that it must be viscous enough, when spread, to lie in a reasonably thick layer upon the wood and subsequently to maintain itself there in a thin but continuous film in spite of pressure. Moreover it must be capable of chilling to a firm jelly very promptly after pressing. It must do this so that it will harden quickly while in position, hold the joint members firmly in place, and commence that long process of drying which will eventually give a continuous film of glue strongly adhering to the wood on either side. This means, in brief, that glue should (1) be of a viscosity low enough to allow penetration of wood structure, yet high enough to prevent excessive loss through seepage and extrusion, and (2) be capable of chilling to a firm jelly at the right temperature. In other words, glue should have a certain moderately high viscosity and jelly strength.

Viscosity and Jelly-Strength Tests

Viscosity and jelly tests of one sort or another have been made ever since people tried to evaluate glue, but it was only recently, the author believes, that any one realized what a direct bearing on the technic of glue handling viscosity and jelly strength really have. These tests originally came into favor for an entirely different reason. Glue users saw in them a chemical rather than a physical significance. Historically it is probable that they owe their importance to the discovery that the deterioration of glue is always accompanied by a lowering in viscosity and jelly strength. Chemists made use of these tests, therefore, to follow the course of the hydrolysis of gelatin and glue. But the author does not see how any relation between viscosity and joint strength could possibly have been worked out prior to the development of a satisfactory joint test, a thing which in his opinion has not been accomplished even yet. The point he is trying to make is that although viscosity and jelly tests have been made for a long time without any definite idea as to their technical significance, they are now amply justified by the new knowledge that the consistency of a glue and its ability to chill at the right temperature are the two very things it is most important to know about it.

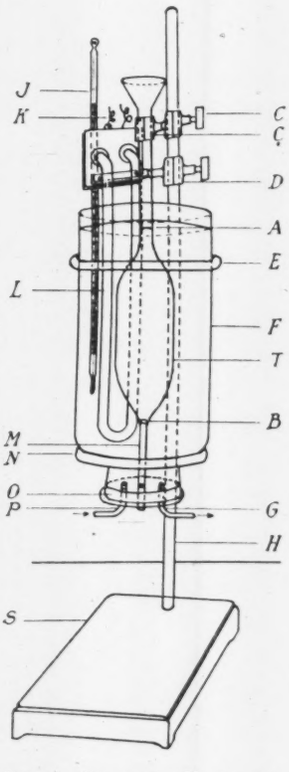

FIG. 1. STANDARD VISCOSITY PIPETTE

A	Upper etched line
B	Lower etched line
C	Clamp supporting pipett
D	Clamp supporting heating element
E	Ring clamp supporting water bath
F	Water bath (5-lb. bottle, bottom cut off)
G	Drain tube
H	Iron support
J	Thermometer
K	Terminals to heating element
L	Heating element
M	Capillary efflux tube
N	Ring supporting water bath
O	Rubber stopper
P	Compressed-air connection for agitation
S	Iron stand
T	Calibrated viscosity pipette

One might think it would be a comparatively simple matter to make a satisfactory viscosity determination on glue, yet any one who has ever attempted it can testify that peculiar difficulties are almost certain to crop up. Glue testing is exceptionally difficult because of complications that are not ordinarily met with. In the testing of other materials the amount of stirring the sample is subjected to usually makes no difference. In glue testing, however, this is not so; it is reported that a glue solution which has once been run through a viscosimeter will not give the same reading thereafter. This is ascribed to the breaking down of its colloidal particles as a result of the mechanical effect of agitation and flow. The thermal history of a glue solution affects its viscosity in the same way. Consequently, the amount of heat a sample receives must be standardized. Temperature effects are responsible for wide variations in the jelly-strength determination, too. These factors, the importance of which was not realized until a few years ago, made it impossible to obtain accurate results with the old methods. Data obtained in different laboratories did not check. This was especially true of jelly-strength data, since every one used a different kind of apparatus and made the determination in a different way. Consequently, no one's results conveyed any definite information to any one else. As a result, therefore, of the inherent difficulties of glue testing and of the strong individualism of the various analysts

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[1] Analytical and Consulting Chemist.

[2] The Gluing of Wood, published by the Forest Products Laboratory, Madison, Wis.; also Glues Used in Airplane Parts, by S. W. Allen and T. R. Truax, National Advisory Committee for Aeronautics, Report No. 66, 1920.

Contributed by the Wood Industries Division for presentation at the Annual Meeting, New York, November 30 to December 4, 1925, of THE AMERICAN SOCIETY OF MECHANICAL ENGINEERS. All papers are subject to revision.

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in the matter of apparatus, methods came to be of small consequence and standard samples became all-important. The reliance that until recently was placed in standard samples was nothing less than a confession of the inadequacy of the testing methods. From 1844 until 1924 glue testing was in the *standard sample stage* of development.

were almost as many different systems of names and grades as there were manufacturers. This gave rise to a multiplicity of grades, names, and designations that was very confusing to the buyer. Viscosity and jelly tests were made, of course, but they were employed merely as a means of comparing unknown glues with some arbitrary standard glue. It is needless to say that while these

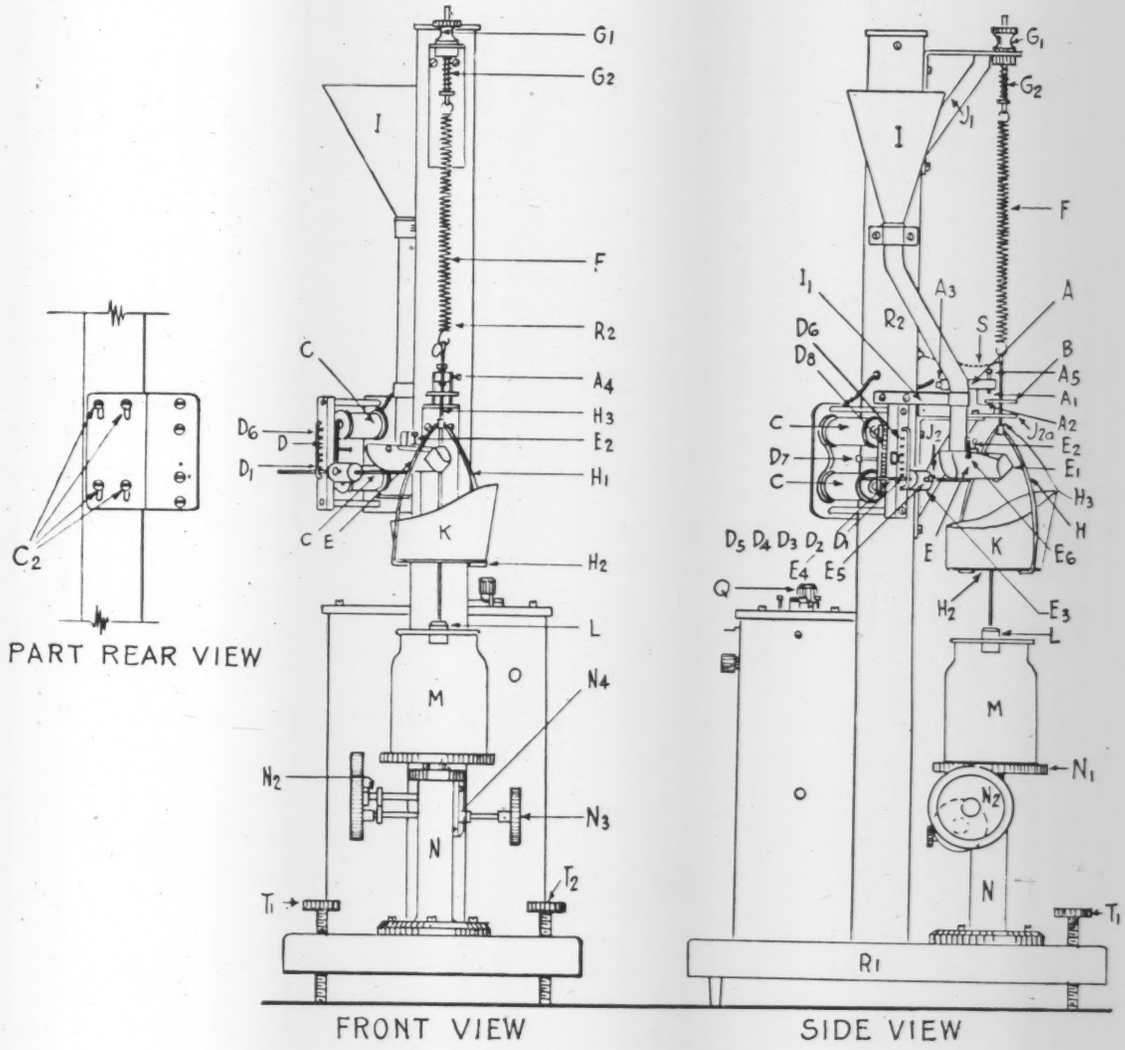

FIG. 2 THE BLOOM GELOMETER

A	Brass contact-point bracket
A_1	Upper contact point
A_2	Lower contact point
A_3	Wood-fiber support for *A*
A_4	Setscrew to hold adjustment screw in position
A_5	Adjustment screw for *A*
B	Pure silver disk (⅜ in. diameter and 1/16 in. thick)
C	Electromagnet
C_2	Adjustment screws for adjusting pitch of clam cut-off *E*
D	Guide bar of automatic shot-control mechanism
D_1, D_2, D_3, D_4, D_5	Lower dog
D_6	Upper dog
D_7	Hairspring coil to keep D_8 in position
D_8	Soft iron bar supporting dogs D_1 to D_6
E	Clamshell spout
E_1	Stationary clamshell jaw
E_2	Adjusting screws to regulate closure
E_3	Weight
E_4	Clamshell arm
E_5	Setscrew to clamp weight to clamshell arm
E_6	Bearing on which cut-off mechanism turns
F	Spiral spring (No 6 steel music wire)
G	Adjustable support for spring *F*
G_1	Thumbscrew nut
G_2	Tension spring
H	Suspended pan and pan arm for shot receiver, etc.
H_1	Pan arms
H_2	Pan
H_3	Rod attached to pan arms supporting disk *B*
I	Shot hopper with delivery tube
I_1	Bracket to hold lower end of shot delivery tube
J_1	Upper supporting bracket attached to frame support R_2
J_2	Lower supporting bracket attached to frame support R_2
J_2a	Guide arm attached to J_2
K	Shot receiver
L	Plunger (12.7 mm. in diameter)
M	Test bottle
N	Elevating platform base
N_1	Platform
N_2	Rack-and-pinion elevating mechanism
N_3	Fine adjustment on rack and pinion
N_4	Brake shoe on adjustment arm of N_3
O	Battery box and batteries
Q	Electrical switch
R_1	Base of gelometer
R_2	Pillar of gelometer
S	Fine-copper-wire coil making contact across from suspended disk to binding post on support
T_1, T_2	Leveling screws

Perhaps it might be well to say just a word more about the era which is now drawing to a close. In 1844 Peter Cooper laid aside a graded set of standard glues and used them thereafter for purposes of comparison.[1] The other glue manufacturers followed his example and devised little systems of their own. Eventually there conditions prevailed, tests were crude and often actually misleading.

It was obvious that glue evaluation could not be put upon a truly scientific basis until testing methods had been improved so as to give results that could be duplicated. If the tests could be refined so that every one examining the same glue would obtain the same results, a twofold object would be attained: viz., glue testing would be raised to the dignity of other kinds of testing, and the primitive system of standard samples could be done away with.

[1] Selection and Testing of Animal Glues for High-Grade Joint Work, by G. M. Hunt and W. L. Jones. Published by the Forest Products Laboratory, Madison, Wis., Mar. 16, 1920.

STANDARD METHODS FOR DETERMINING VISCOSITY AND JELLY STRENGTH

This has now been accomplished, thanks to the very able work of the chemists of the National Association of Glue Manufacturers. The author will not attempt any lengthy discussion of the new methods, but will refer the reader to an article entitled, Standard Methods for Determining Viscosity and Jelly Strength of Glue, which appeared in *Industrial & Engineering Chemistry*, March, 1924, page 310, and which gives the methods adopted by the Association in October, 1923. Briefly stated, viscosity is determined in a glass pipette surrounded by a water bath as shown in Fig. 1. The pipette is standardized with the aid of liquids of known viscosity, so that the *absolute viscosity* of an unknown glue can be calculated from the experimental data. The procedures of weighing, soaking, heating, stirring, etc., are described in such detail that every factor of importance is taken into account. The only criticism the author can think of is that the moisture content of the sample, which obviously must be a factor, is neglected. That is a matter which it might be well to rectify in the future.

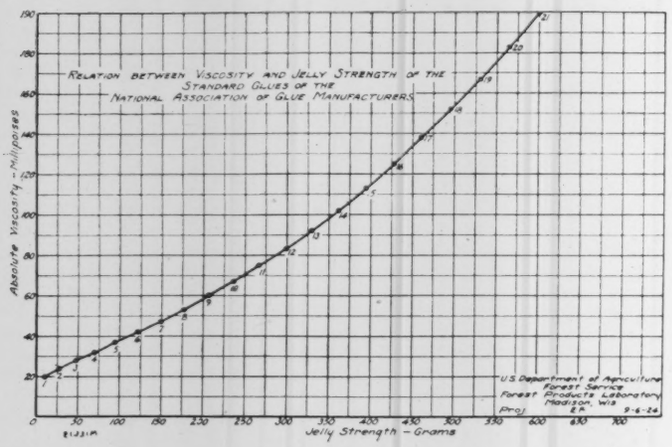
Relation between Viscosity and Jelly Strength of the Standard Glues of the National Association of Glue Manufacturers
Absolute Viscosity - Millipoises
Jelly Strength - Grams
U.S. Department of Agriculture
Forest Service
Forest Products Laboratory
Madison, Wis.
Proj. E.R. 9-6-24

FIG. 3 RELATION BETWEEN VISCOSITY AND JELLY STRENGTH OF STANDARD GLUES

With this one exception there is nothing but praise for the N.A.G.M. method. It gives results that check better than the author had ever thought possible. He has also found it quick and convenient and can make viscosity determination with an N.A.G.M. pipette in 15 minutes, whereas with the old Engler viscosimeter he was formerly unable to make a complete determination in much less than 45 minutes.

The Bloom gelometer, described in the same article and shown in Fig. 2, is so far in advance of all other jelly testers that there is no comparison. It is the first practical jelly tester to yield results that can be checked. It is rapid and automatic, and the personal element is rendered almost negligible. Another good feature of the new jelly test is that it is made on the solution remaining from the viscosity determination. This obviates the need of weighing out a special solution for the jelly test. There is one thing, however, that every one who contemplates using the Bloom gelometer for jelly testing should know: it is so exceedingly sensitive that it is necessary to keep the jellies at a temperature of exactly 10 deg. cent. If the sample is not kept constant within 0.1 deg., the results will vary widely. There is no thermostat on the market that will serve the purpose, and analysts have learned that it is no easy matter to construct an ice chest that will maintain a constant temperature of 10 deg. cent. It takes engineering skill and no small amount of money to make a suitable thermostatic ice chest. This fact, the author believes, is the greatest obstacle to a general adoption of the method.

The two methods just mentioned have been tried out for some time and found entirely satisfactory. The benefit that will accrue to the public from the adoption of these methods is nothing less than the abolition of standard glues. Glues will henceforth be described in terms of their constants: namely, their *absolute viscosity* in millipoises and their jelly strength in grams. Of course there will be standards in the new order of things, but they will be of a different kind, for they will merely be names to designate certain combinations of viscosity and jelly-strength values. For instance, a glue having a viscosity of 92 millipoises and a jelly strength of 331 grams will be known as a No. 13 glue (see Fig. 3). There will be 21 such grades, ranging from the top to the bottom of the scale. The combinations of viscosity and jelly-strength values that constitute these 21 standard grades are shown in Fig. 3.

A circumstance that detracts from the value of these standards is this: that not all glues fall exactly on the normal jelly-viscosity curve. Oftentimes a glue that is a No. 14 in viscosity is a No. 12 in jelly strength. Such a glue is not rightly a No. 14 nor a No. 12, nor is it a No. 13. It is spoken of, therefore, as being of such-and-such a viscosity and of such-and-such a jelly strength. This means that the tendency is to describe a glue in terms of its physical constants rather than in terms of its conformity to some more or less arbitrary standard glue. The time is probably coming when bids will be taken on a glue of so many millipoises rather than on a glue of a certain grade.

The National Association of Glue Manufacturers comprises the majority of large manufacturing firms. The Peter Cooper Glue Company does not belong, and perhaps there are a few others. Nevertheless, even though there are a few outsiders, a great simplification has taken place, and the glue user stands to profit by it. At most we now have but two systems of grades to deal with, whereas formerly we had a great many more.

The business of glue buying is changing, as it were, overnight. With our new precision instruments, with our ability to define glues in terms of physical constants, and with our reduction of grading systems to two, we are in a fair way to rid glue of all the mystery and hocus pocus that formerly made glue buying one of the most perplexing tasks of the woodworker.

Centrifugal Compressors for Diesel Engines

Installations of Centrifugal Compressors for Scavenging and Supercharging—Centrifugal Compressors for Diesel Engines—Advantages of Centrifugal Compressors for Two-Cycle Scavenging—Supercharging Experiments—Theory of Supercharging a Diesel Engine—Exhaust-Gas Turbine for Scavenging and Supercharging

By SANFORD A. MOSS,[1] WEST LYNN, MASS.

DIESEL engines are well established, as far as thermal efficiency is concerned, and most builders are now making efforts to secure a more favorable situation with regard to power output for a given weight and cost. One development in this direction, which is proceeding quite rapidly, is the use of high-speed centrifugal compressors for scavenging and supercharging both two-cycle and four-cycle engines. The use of high rotative speeds has decreased to a remarkable extent the weight and cost of many types of apparatus, and the use of centrifugal compressors gives some of this advantage to the Diesel engine.

The centrifugal compressor consists of an impeller carrying properly shaped blades, which rotates at high speed in a casing with suitable inlet and discharge conduits. There is appreciable clearance, so that no rubbing or sliding occurs. Such compressors are now in extensive commercial use, both in this country and abroad, for supplying air for various purposes. Types which supply air at pressures of from 1 to 5 lb. per sq. in. are being used commercially with Diesel engines.

The two-cycle Diesel engine, which has only an expansion stroke and a compression stroke, requires that air at a pressure of from 1 to 3 lb. per sq. in. be supplied by a scavenger of some type. This air is introduced into the cylinder through appropriate valves or ports when the piston is near the end of the expansion stroke, in such a way as to sweep out the exhaust gases through cylinder wall ports uncovered by the piston, and to fill the cylinder with a new charge.

Two-cycle scavenging is accomplished in four ways. The original "Crankcase Method" compressed the air in the crankcase by means of the lower side of the piston, but more recently there has been used a separate piston and cylinder operating in synchronism with the main piston.

The second or "Timed Method" uses a reciprocating blower not in exact unison with the power piston, but with such timing relative to it that the delivery of the air corresponds to the opening of the scavenging port.

The third or "Receiver Method" uses a reciprocating blower and a large receiver within which is an approximation to constant pressure. Claims have been made that a reciprocating blower with timing gives better performance than the constant pressure of a receiver, but it is nevertheless true that many engines with constant-pressure scavenging are as successful as those with timing. It is, however, difficult to secure a receiver large enough to give a really constant pressure with a reciprocating blower, and there is always an appreciable fluctuation of pressure. Hence, in order to secure proper scavenging the minimum pressure must have a required value.

The "Centrifugal-Compressor Method" of scavenging give a constant pressure. Tests show that the pressure actually needed for given results is lower than the average value of the irregular pressure of the Receiver Method.

Supercharging of four-cycle engines is desirable because of the fact that without it the charge of air at the beginning of the compression stroke has a low density, due to a lower pressure and a higher temperature than that of the atmosphere. Supercharging gives an increase in the amount of charge, and hence in the amount of power delivered, by supplying the intake manifold with air at a pressure appreciably above atmospheric.

By increasing the pressure of the scavenger of a two-cycle engine, any desired amount of supercharging may also be obtained, provided that the air ports remain open after the exhaust ports have closed or that the engine exhausts against a back pressure.

Installations of Centrifugal Compressors for Scavenging and Supercharging

A great many installations of centrifugal compressors have been made by the Brown-Boveri Company, of Switzerland, C. H. Jaeger, of Leipzig, the British Thomson-Houston Company, and other European builders. Most of these are driven by high-speed direct-current motors and are for marine use. Fig. 1 shows a typical installation with a Sulzer Brothers Diesel Engine[1] and Fig. 2 shows a Brown-Boveri machine.[2]

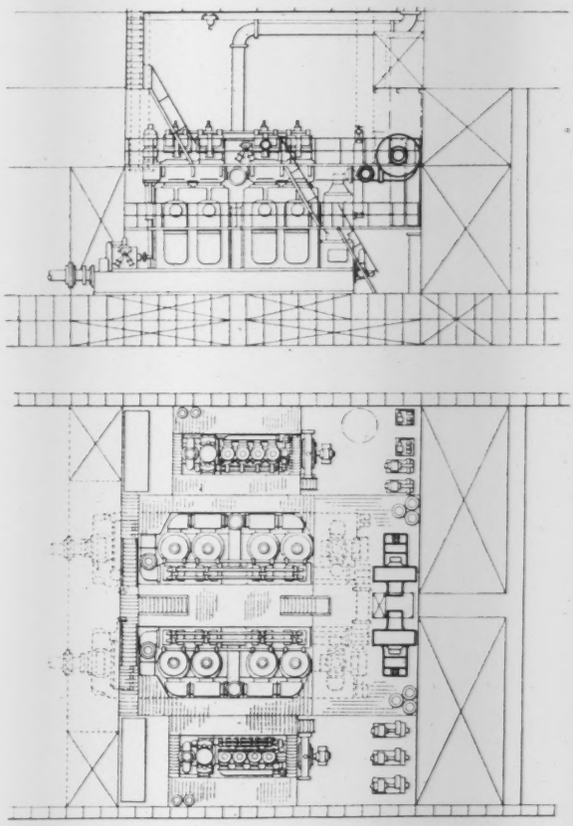

Fig. 1 Layout of Sulzer Two-Cycle Marine Diesel Engine with Centrifugal Compressor for Scavenging

Figs. 3 and 4 show one of two identical installations of General Electric Company centrifugal compressors supplying scavenging air for Sun-Doxford two-cycle engines in the ore-carrying vessels *Benson Ford* and *Henry Ford, 2nd*, operated by the Ford Motor Co., of Detroit.[3]

Each of these sets consists of a direct-current motor rated at 180 hp., 1150 r.p.m., and 230 volts, and a centrifugal compressor rated at 12,000 cu. ft. per min., 2 lb. per sq. in., 3600 r.p.m., with herringbone step-up gears between. This allows the use of a standard motor at a usual speed.

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[1] Engineer, Mechanical Research Department, General Electric Company. Mem. A.S.M.E.

For presentation at the Annual Meeting, New York, November 30 to December 4, 1925, of The American Society of Mechanical Engineers. All papers are subject to revision.

[1] From the *Brown-Boveri Review*, April, 1924, which gives an excellent description of the use of centrifugal compressors for scavenging.

[2] From *Motorship* (New York), March, 1923.

[3] A complete description is given in *Motorship* (N. Y.), Sept., 1924.

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The most outstanding application of a two-cycle engine with a centrifugal scavenger is on the *Aorangi*,[1] a quadruple-screw liner of 23,000 tons displacement. Four 6-cylinder, two-cycle Sulzer engines developing a total of 13,000 shaft hp. at 127 r.p.m. give the greatest power ever used in a Diesel ship.

The scavenging and supercharging air is supplied by two centrifugal blowers, each capable of delivering 34,000 cu. ft. of air per minute against a pressure of 1.75–2.1 lb. per sq. in., while a third

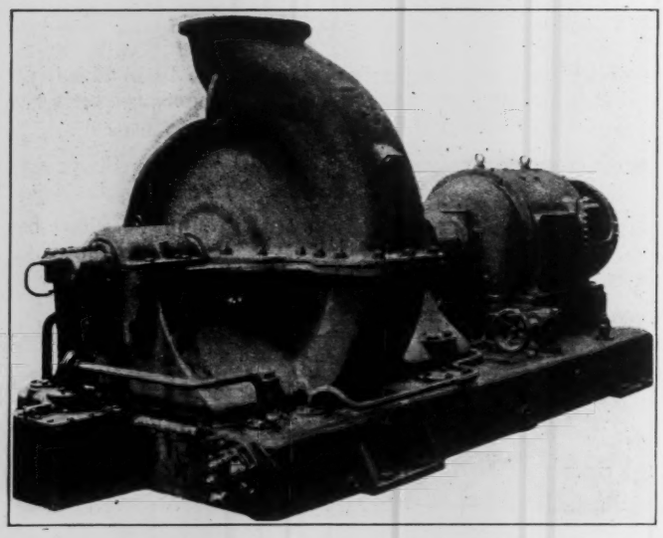

FIG. 2 BROWN-BOVERI SUPERCHARGER FOR THE FOUR-CYCLE MOTORSHIP "MONTE SARMIENTO"

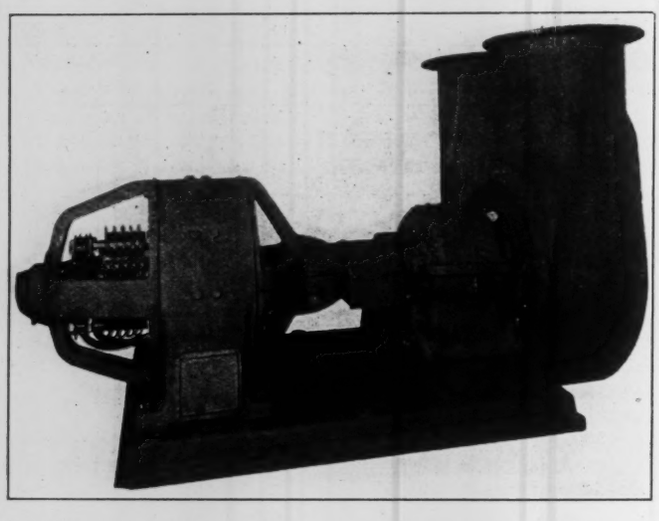

FIG. 3 GENERAL ELECTRIC CENTRIFUGAL COMPRESSOR FOR SCAVENGING TWO-CYCLE DIESEL ENGINES ON FORD ORE VESSELS

unit acts as a stand-by. Besides the usual scavenging ports uncovered by the stroke of the piston, there is a row of automatic multiple-disk valves just above them, which open when the exhaust pressure falls below that of the scavenging air and which remain open after the exhaust valves have been closed on the return stroke, until the air pressure in the cylinder equals that of the scavenging air. This provides for greater than atmospheric pressure in the cylinder at the beginning of compression, or true supercharging.

Another application in an entirely different field is that of the new Baldwin locomotive, whose driving element is a Knudsen Diesel engine scavenged and supercharged by a General Electric Company centrifugal compressor. (See Fig. 5.)

The Knudsen Diesel engine is of the two-cycle, inverted-V type, corresponding to an opposed-piston engine with the cylinder divided in the middle to form an acute angle so that the pistons are operated from separate crankshafts which are geared together to maintain the required synchronism and to provide a common driving shaft. The compressor supplies air at from $1\frac{1}{2}$ to 3 lb. per sq. in., whereby scavenging and supercharging from 10 per cent to 30 per cent are obtained, according to the speed and power requirements.

In England a number of installations have been made with centrifugal compressors for two-cycle scavenging driven by steam turbines.

The greatest use of centrifugal compressors for supercharging four-cycle engines has been in Germany, and they are now used there on practically all new projects.

The supercharged four-cycle Diesel engine which has been in operation for the longest time is that of the Norwegian ship *Tiradentes*. When the ship was first operated it did not develop sufficient speed, and a supercharger was added which increased the indicated hp. from 3100 to 3600 and the speed of the ship from $10\frac{1}{2}$ knots to $11\frac{1}{2}$ knots.[1]

The engines were built by the Allgemeine Elektricitäts Gesellschaft, of Germany, under license from Burmeister & Wain. In order to avoid increase of compression pressure with supercharging,

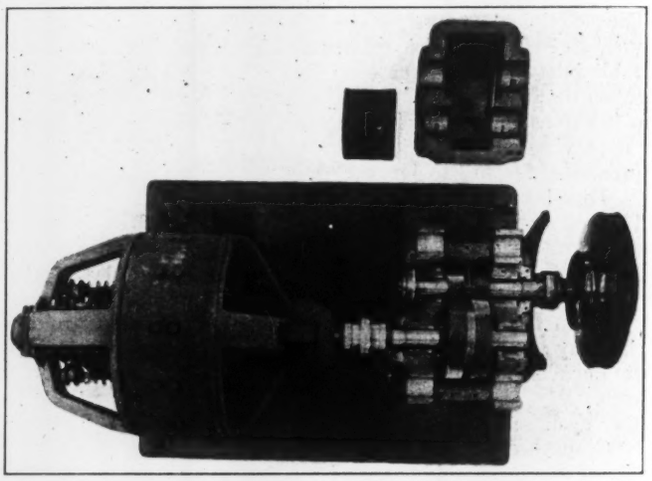

FIG. 4 INTERIOR OF GENERAL ELECTRIC CENTRIFUGAL COMPRESSOR SHOWN IN FIG. 3

the clearance space was increased by lowering the position of the pistons about 0.2 in.

The supercharger is a centrifugal compressor driven by a direct-current motor, built by the Brown-Boveri Company and similar to their scavenging compressors already described. The supercharging pressure is only 0.85 lb. per sq. in., which requires an electrical input of 33 kw.

With the supercharger in operation, the exhaust temperature varies from 806 deg. to 842 deg., which is higher than with ordinary four-cycle engines. The exhaust is entirely invisible and there are no soot deposits in the exhaust conduit. Every 40 days the exhaust valves have to be reground instead of every 60 days as would ordinarily be the case, and a somewhat increased amount of cooling water is required. These points are regarded as a very small price to pay for the increased power.

Since the *Tiradentes* installation a number of other four-cycle Diesel-engined ships have been supercharged in practically the same way. Among these may be mentioned the *Tampa* (see Fig. 6), *Tungsha*, *Taiwan*, and *Tortugas*,[2] belonging to the Wilhelmson Company as does the *Tiradentes*. Some of these vessels frequently visit the United States. Two 14,000-ton Diesel passenger liners have recently been completed in Germany, with superchargers as an integral part of the design. These are the *Monte Sarmiento*[3] (see Fig. 2), and the *Monte Olivia*. A number of other supercharged ships are also in the process of construction or have just been completed.[4]

[1] *Jour. Am. Soc. Naval Engrs.*, Feb., 1925, p. 159; *Motorship* (N. Y.), Mar., 1925, p. 189; *The Engineer*, p. 111, Jan. 23, 1925, p. 111.

[1] A full description of the installation is given in *Motorship* (N. Y.), Mar., 1921, p. 179.

[2] Scholz, *Werft, Reederei, Hafen*, June 22, 1925, p. 260; *Motorship*, London, Nov., 1924, pp. 291 and 294.

[3] *Motorship* (N. Y.), Dec., 1924, p. 308, and Feb., 1925.

[4] *Motorship* (London), Nov., 1924, p. 294.

Although the matter of supercharging Diesel engines has only recently begun to attract attention, the proposition is really quite old and many of the early workers on both Otto-cycle and Diesel-cycle engines mention it. Propositions were also made years ago to use separate blowers with Diesel engines in mountainous regions, in order to bring the power to the value obtained near sea level.

CENTRIFUGAL COMPRESSORS FOR DIESEL ENGINES

The Centrifugal Compressor resembles in a general way the well-known fan blower. However, the details of design of the conduits in the casing leading to and from the impeller, and of the vanes on the impeller, are attended to more carefully and the rotative speeds are much higher, varying from 3000 to 10,000 r.p.m. The speed of the air through the apparatus is also comparatively high. For these reasons the vanes and passages must be designed to handle the air without shock. The air leaving the impeller possesses a velocity of the order of magnitude of the peripheral speed of the impeller, which represents an amount of energy equal to about one-half the power output of the driver. There is a collecting passage or diffuser beyond the impeller which receives this air and decreases its velocity without shock, so that the energy represented by the velocity is converted into energy represented by pressure. Due to the high speed and general design, centrifugal compressors are comparatively light and compact.

Centrifugal compressors have been in competition with older types of compressors for many years and their use is rapidly increasing. Many actual comparisons have been made of efficiencies of the several types, and these have shown that a centrifugal compressor and good compressors of the older types have efficiencies which are essentially equal when both are in good condition. A centrifugal compressor is subject to no deterioration under usual circumstances, and hence maintains its original efficiency. The older compressors always involve rubbing or sliding parts. In some cases these are maintained at appreciable expense so as to retain the initial efficiency. With average maintenance there is soon enough leakage to materially decrease the efficiency.

Drivers for centrifugal compressors for Diesel engines are often direct-current motors. In case of installations on board ship this is usually desirable for the reason that there is an electric generator driven by an auxiliary engine which supplies power for winches and other auxiliary machinery in port, and which therefore gives capacity for supplying compressor motors at sea. The European sets have usually been directly connected to especially designed high-speed motors running at from 2500 to 4500 r.p.m. The American sets have usually used electric motors with lower speeds and centrifugal compressors with higher speeds, with gearing between. This has given the possibility of selecting the speed of each unit in the most advantageous way and without compromise for the benefit of the other. The low-speed electric motor avoids difficulty of commutating, etc., and the high-speed compressor makes cost, weight, and space a minimum.

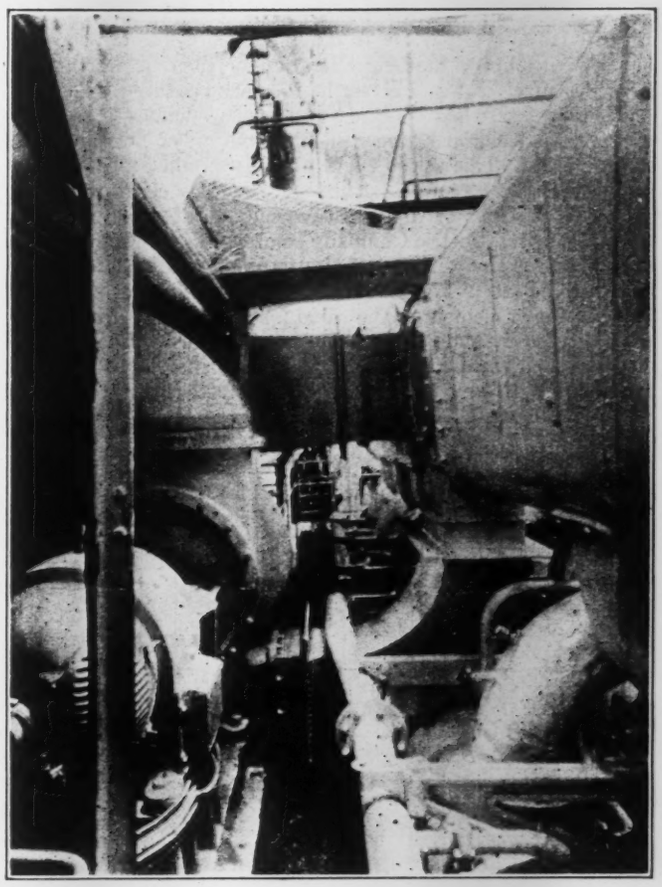

FIG. 6 SUPERCHARGING COMPRESSOR AND RECEIVER ON MOTORSHIP "TAMPA" (FOUR-CYCLE ENGINE)

In some cases, as in the Knudsen locomotive previously described, a centrifugal compressor for supercharging or scavenging is directly geared to the engine. In such a case, and if the engine conditions are such as to give a proper volumetric efficiency, there is automatic pressure compensation for the variation of friction through the passages with engine speed, and if there is supercharging in addition, this automatically varies as the square of the engine speed. So far as is known, this is the only use thus far made of a geared supercharger on a Diesel engine. There have been made, however, a few installations of geared superchargers on Otto-cycle engines for racing automobiles by the author and others. A great many experiments on geared superchargers for Otto-cycle airplane engines have been made, but no commercial use has occurred as yet.

In Great Britain certain maritime rules require that passenger ships have a number of steam-driven pumps and steam for heating the ship, etc. Hence most British ships with centrifugal compressors for Diesel engines have had steam turbines as drivers.

There have not been enough installations of separate blowers for stationary Diesel engines to indicate the type of driver which would be most usual. It would be possible to drive the compressors direct by alternating-current induction motors at 3600 r.p.m. synchronous speed. The General Electric Company is now supplying a great many such alternating-current sets for miscellaneous industrial purposes.

The use of an exhaust-gas turbine is also possible with both Diesel-engine scavenging and supercharging, as discussed later.

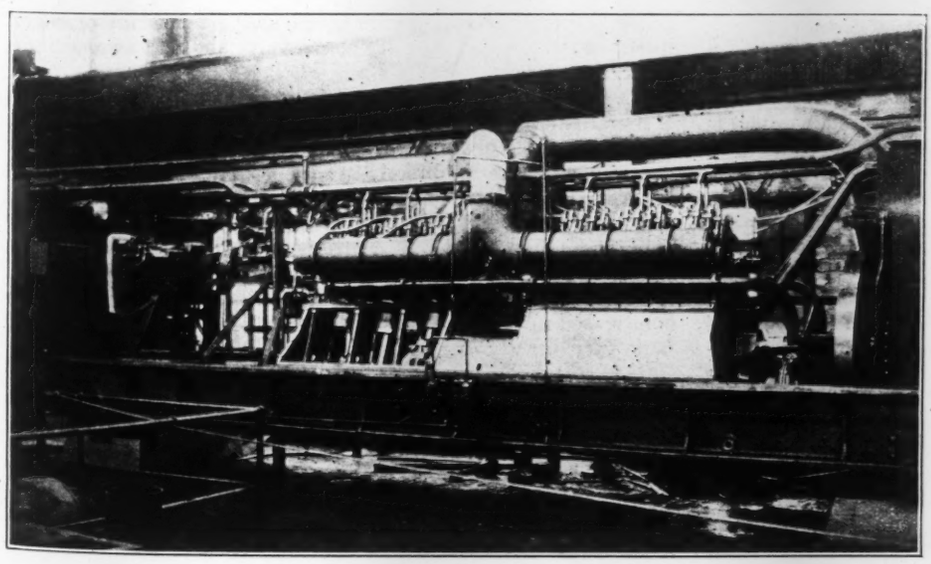

FIG. 5 BALDWIN LOCOMOTIVE WITH KNUDSEN TWO-CYCLE DIESEL ENGINE AND GENERAL ELECTRIC SCAVENGER AND SUPERCHARGER

Centrifugal-Compressor Characteristics. The characteristics of a centrifugal compressor are so different from those of a positive-displacement blower that consideration of them is worth while. The upper curve of Fig. 7 gives the pressure produced by a centrifugal compressor when operated at various volumes. As will be seen, there is a comparatively slight variation of pressure as the volume varies. This is the curve actually produced by the Ford compressors, Figs. 3 and 4. This curve is for a constant position of the field rheostat of the direct-current motor which drives the set. All General Electric Company compressors give similar curves whether driven by direct- or alternating-current motors or by steam-turbine sets. Fig. 12 shows a sectional view of such a set. The compressor element is the same for any type of driver.

In the case of a direct-current motor there is usually a field rheostat which can be set at various positions so as to give speed variation. With any one setting of this field rheostat the speed decreases somewhat as the load increases, as shown by the lower curve of Fig. 7.

With the field rheostat in one extreme position, so that there is the "full field" on the motor with no rheostat resistance in series

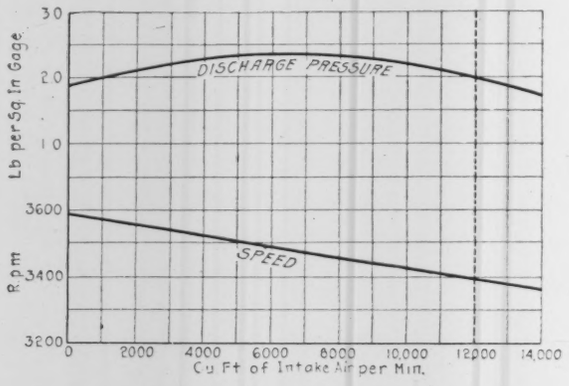

FIG. 7 DISCHARGE PRESSURES AND SPEEDS OF GENERAL ELECTRIC CENTRIFUGAL COMPRESSORS FOR FORD ORE VESSELS
(Based on air at 14.4 lb. per sq. in. abs. and 60 deg. fahr.)

with the field coils, there is obtained the minimum speed and lowest pressure. This should be so selected as to be the value required during starting or low-speed operation of the engine. By turning the field rheostat by hand, resistance is added to the field circuit and the field is "weakened," which increases the speed. The maximum speed and weakest field of the motor should be so selected that the corresponding pressure has the greatest value which is desired. In an ordinary motor a pressure variation of about 15 per cent may be obtained in this way by the use of a field rheostat. With special arrangements, greater variations may be obtained.

When a centrifugal compressor is applied to any ordinary industrial use, such as supplying air for burning gas or oil, it is connected up to a piping system. In this system there are a certain number of openings, such as nozzles in burners, through which is discharged the air which the machine supplies for the purpose in hand. The pressure existing within the piping system forces a definite amount of air through the given burner openings and thus establishes the load or volume in cubic feet per minute which the compressor must supply. This load is increased or decreased as the number of openings in oil burners or the like is varied to suit the amount of work that is required of the machine. The operation in this respect is the direct antithesis of the operation of a positive-displacement machine such as a Roots blower or a reciprocating piston compressor. The volume passing through a displacement machine is directly fixed by the displacement regardless of the openings in the piping system, whereas the volume passing through a centrifugal machine is fixed entirely independently of its speed by the number of openings which are arbitrarily introduced into the piping system. In other words, the centrifugal compressor gives a practically constant pressure while automatically adjusting itself to the volume demanded by the number of burners or other openings required at the time.

While the pressure produced by the machine is nearly constant, regardless of the volume, this is only true within certain limits. The exact pressure when a given volume is made to flow through the compressor by a given number of openings in burners or the like, is given by a characteristic curve such as in Fig. 7. For very large volumes the pressure is somewhat below that at the rated volume. However, if a machine is properly selected for the work in hand, whether for a Diesel engine or for oil burning, the pressure will be practically constant for all ordinary variations of volume. Unfortunately, however, cases sometimes arise where machines are not properly selected. The volume imposed on a machine by the work in hand may then be very much less than that for which the machine is suited, in which case there are the pulsations alluded to later; or the volume may be much greater, in which case the pressure delivered by the machine is low and the power is high.

When a centrifugal compressor of such characteristics is applied to scavenging or supercharging a two-cycle or four-cycle engine, the operation is essentially as described above if the valve and port arrangements and engine speed are such that enough air flows into the cylinder at each charge to bring the pressure within the cylinder up to a value approaching that in the discharge pipe of the compressor. Under such circumstances the cylinder displacement is practically filled at each stroke, and the total load on the compressor is directly given by the total displacement of the engine.

It must be understood that the engine displacement is never quite filled with air at atmospheric temperature and at the pressure existing within the compressor discharge pipe. The temperature within the cylinder is somewhat higher and the pressure somewhat lower than these values. Hence the actual volume of air is 80 to 90 per cent of the volume given by the direct displacement, and the ratio is called "volumetric efficiency." Values of this factor have been determined for some Otto-cycle engines and range from 85 to 88 per cent. Values for Diesel engines are at rated speed undoubtedly nearly the same if the speed and valve and port arrangements are proper.

The above statements apply directly to a four-cycle engine or to a two-cycle engine with a special inlet valve which remains open after the exhaust valve is closed, or to a two-cycle engine running at a speed exactly suited to the port areas, so that the exhaust gases are completely and exactly displaced without discharge of air through the exhaust port. In the extreme case of a high-speed two-cycle engine without a special inlet valve there is no such thing as volumetric efficiency, and the volume of air required is independent of the engine speed. This is because during the entire time that the inlet port is uncovered by the piston, there is flow of air through it, the amount of which is determined only by the pressure in the air conduit and the fraction of the time which the inlet port is opened. When the engine speed is varied, the total time during which the inlet port is opened is unchanged, so that the volume of air is unchanged. There are, of course, many intermediate conditions between the two extreme conditions above specified—volume of air in proportion to engine speed and independent of engine speed—which depend upon the actual valve and port arrangements in any given case.

When a centrifugal compressor is operated in the neighborhood of its rated load, if the compressor speed is varied the pressure rise varies approximately as the square of the speed. The rated volume of a compressor varies directly as the speed, hence if the speed of a given compressor is doubled, the volume to which it is suited is also doubled and the pressure rise which it produces at the double volume is quadrupled. A centrifugal compressor geared to an engine is therefore automatically maintained at its rated volume, since as its speed is increased the volume demanded by the engine is increased in the same ratio. Friction losses through valves and ports vary directly with the square of the volume; hence, as already stated, a geared compressor automatically gives a pressure rise which overcomes friction losses at all speeds, and, if there is supercharging, gives supercharging pressure varying with the square of the speed.

A centrifugal compressor serving a Diesel engine should always have a butterfly or other type of gate valve at the compressor inlet. The compressor is started with the inlet valve closed before the engine is started, and operated at the lowest speed possible by using the field rheostat or the like. The Diesel engine is then started, and as it attains speed the compressor inlet valve is gradually

opened. The valve is opened completely if the full pressure of the compressor is desired, or partially if the full pressure is not desired. After the inlet valve is opened completely with the compressor at the lowest speed possible, the speed is increased by means of the field rheostat if necessary. When three centrifugal compressors are arranged so as to serve two engines, additional butterfly or other gate valves are desirable at discharge as well as at inlet.

As will be seen from the characteristic curve of Fig. 7, when the load on a centrifugal compressor is very much below normal, the pressure rises with increase of load. This gives an unstable characteristic and therefore a centrifugal compressor operating at a load very much below normal or at no load such as when the compressor is running with discharge gate open before the Diesel engine is started, undergoes so-called "pulsations" or pressure fluctuations accompanied by a back rush of air out of the inlet and a peculiar noise. This is eliminated by throttling or completely closing a valve at the inlet, and hence centrifugal compressors are always controlled by such a valve during starting or light-load operation as already specified.

Every Diesel installation has some sort of a high-pressure air compressor for starting and maneuvering purposes, and many have air compressors for the injection of fuel. Recent installations of scavengers or superchargers have been so arranged that they supply these compressors also. This makes it possible to use a smaller compressor for a given amount of air or gives a larger amount of air from the same compressor. A scavenger for 2 lb. pressure will increase the compressor capacity about 15 per cent. Centrifugal superchargers and scavengers are always provided with inlet conduits so that air supply can be drawn from outside the engine room. It has been proposed to connect these conduits to parts of the ship which need to be ventilated so as to take the place of ventilating blowers which would otherwise be required.

Advantages of Centrifugal Compressors for Two-Cycle Scavenging

Weights and Dimensions of Reciprocating Blowers and Centrifugal Compressors. The most important advantage of scavenging a two-cycle engine by a separate centrifugal compressor is the great reduction of cost and weight. See Fig. 8. When a Diesel engine is designed for a separate air supply, besides dispensing with the huge blowing cylinder with all its accessory valves, rods, etc., the massive base can be made considerably shorter and more than proportionately lighter.

A comparison of the weights and dimensions of the Ford Diesels, which have separate blowers, with the engines of the ship *Challenger* which are of similar design but with reciprocating scavengers, was made by the Sun Shipbuilding and Dry Dock Company of Chester, Pa., who built both engines. This shows that the weight has been reduced from 275 lb. per b.hp. to 246 lb. per b.hp., including the spare compressor set, which is a net saving of 36,000 lb. or 4.4 per cent. The length was also calculated to have been reduced 8 ft. or 13 per cent. The use of old patterns in the Ford engines did not permit of the full reduction possible with the centrifugal-compressor system.

Improvement in Low-Speed Operation has been obtained with centrifugal scavengers due to increased supply of scavenging air. This is due to use of full pressure with decreased air friction resulting from the low speed. Reports from abroad and from the two installations in the United States already alluded to, uniformly praise centrifugal-compressor scavengers for giving ease in maneuvering. The starting of a Diesel engine is also facilitated since the centrifugal compressor, being started in advance, gives a full supply of warm air for the first firing stroke.

Efficiency. No data are available as to the actual efficiency of reciprocating scavengers, but efficiencies greater than 75 per cent are actually guaranteed for centrifugal compressors. The efficiency of a reciprocating blower is probably lower on account of the losses in volumetric efficiency and of friction due to driving a low-pressure blowing piston from the heavy crankshaft of a Diesel engine. However, it is customary to assume that such losses are about equal to the losses in the electric generator and in the motor and the centrifugal compressor. On the basis of this assumption the relative power taken from the crankshaft is directly proportional to the pressure rise. It has already been pointed out that the pressure rise necessary to secure a given quantity of scavenging is somewhat less with a centrifugal compressor than with a reciprocating blower, so that the power would be reduced in practically the same proportion. This conservative estimate shows the power for scavenging a centrifugal compressor to be about 70 per cent of that for a reciprocating blower.

Incidental Advantages. If an accident happens to a reciprocating scavenger the entire plant is put out of commission, which, if it should occur to the main engine at sea, would make the vessel helpless until repaired. On account of the low cost, most installations of centrifugal compressors are made in duplicate. This gives an insurance against accident which is not possible with reciprocating scavengers.

The valves and piston rings of a reciprocating scavenger must, of course, be carefully kept in order to keep up the initial scavenging pressure and to maintain the engine power. This maintenance is entirely eliminated with a centrifugal scavenger. A margin in the amount of scavenging air is provided for with but little additional cost in the case of a centrifugal scavenger. The size and

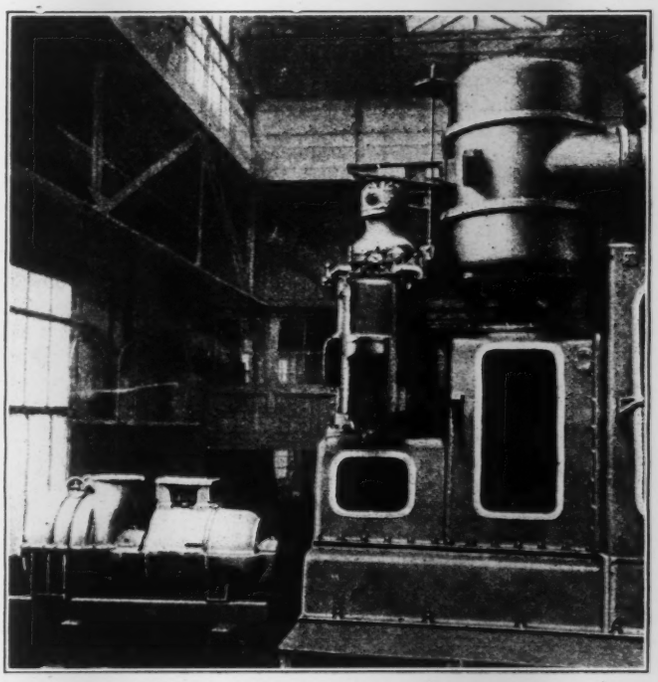

FIG. 8 VIEW SHOWING COMPARATIVE SIZE OF A TWO-CYCLE RECIPROCATING SCAVENGING BLOWER AND A CENTRIFUGAL COMPRESSOR OF THE SAME CAPACITY

weight of a preciprocating scavenger make builders loath to provide such a margin with it. The extra scavenging air thus possible gives cleaner exhaust and less deposits in cylinder and exhaust conduit, as well as a possibility of a slight increase of power.

Size of Centrifugal Compressors for Two-Cycle Scavencing

The displacement of a reciprocating scavenger is usually selected on the basis of an increase of about 25 per cent over the displacement of the main engine. This gives no information as to the volume required of a centrifugal scavenger because of the great uncertainty regarding the air actually delivered by a reciprocating machine. Some foreign builders of centrifugal scavengers have found it desirable to use diffusers which can be adjusted to suit the volume actually required when the apparatus is installed in connection with the engine. Approximate computations of the air required from the centrifugal compressor are as follows: The displacement of the reciprocating blower of the ship *Challenger* corresponds to 4 cu. ft. per hp. A volumetric efficiency of about 87.5 per cent gives a round figure of 3.5 cu. ft. of actual air per b.hp. required from a centrifugal scavenger, and this figure has been used by some builders. Computations made on the basis of the displacement of the main engine are somewhat uncertain owing to uncertainty as to the exact amount of exhaust gas scavenged out. This may be

less than the displacement, equal to the displacement, equal to the displacement plus the clearance volume, or more than this so as to force some air through the exhaust port. The flat pressure curve of the centrifugal compressor gives little difficulty as far as delivery of air is concerned, and the compressor operates well even though it is somewhat too small or too large. Too large a motor as driver also gives no difficulty, which leaves the possibility of an overload on the motor as a principal item to be considered. So much depends upon the engine design and the scavenging pressure as to make the selection of the exact size of the centrifugal compressor more or less of an empirical matter, to be based on experience which has been obtained with each engine design.

Supercharging Experiments

A great many experiments have been made in which two-cycle and four-cycle engines have been supercharged, and an appreciable gain in power found.[1]

These experiments involved supercharging to two quite different degrees. In some the amount of supercharging has been comparatively slight so as to mainly overcome the friction drop through inlet conduits and valves. In others a much greater amount of supercharging has been used. The experiments have been mainly with four-cycle engines.

A slight amount of supercharging with blowers delivering air at a pressure of 400 to 600 millimeters of water, or 0.75 lb. per sq. in. has been found to give increases of power much greater than proportional to increased charge in the displaced volume, the reasons for which are discussed later. This type of supercharging is the only one which has come into actual commercial use at the present time. A large amount of supercharging varying from 1.5 lb. per sq. in. to 30 lb. per sq. in. has been used in other experiments and the net power obtained actually, or computed after allowing for power for driving the supercharger. In all cases there was a gain in this net power, even though it involved an increase in the pressure at the end of expansion. A number of experiments have been

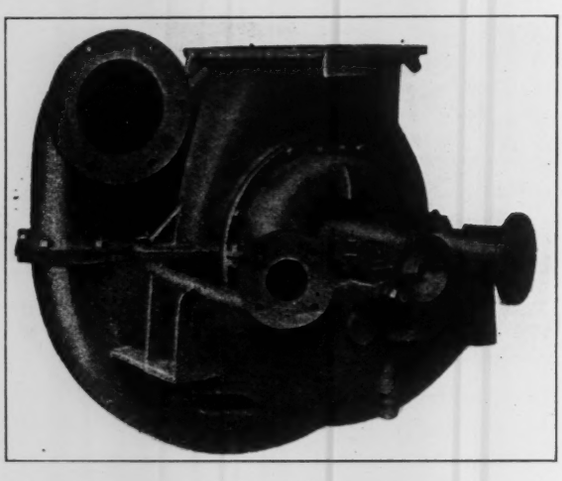

Fig. 9 Brown-Boveri Diesel-Engine Exhaust-Gas Turbine

published by Rateau and by the author on the supercharging of an Otto-cycle engine by means of an exhaust-gas turbine for use on airplanes operating at high altitude, but while the use of such an exhaust-gas turbine has often been proposed for a four-cycle Diesel engine, no experiments have ever been published. It is understood that a number of tests were made on a European locomotive some years ago. A great many airplanes with four-cycle Otto engines and exhaust-gas turbines, have been supplied for commercial use by Rateau in France, and by the General Electric Company in this country. The Brown-Boveri Company have advertised that they will supply similar exhaust-gas turbines for Diesel engines. See Fig. 9. So far as is known, however, no commercial installation of an exhaust-gas turbine on a Diesel engine has ever been made.

Some unpublished experiments have recently been made in the United States, in which an opposed-piston two-cycle engine has been supercharged with initial pressures from 1½ to 30 lb. per sq.

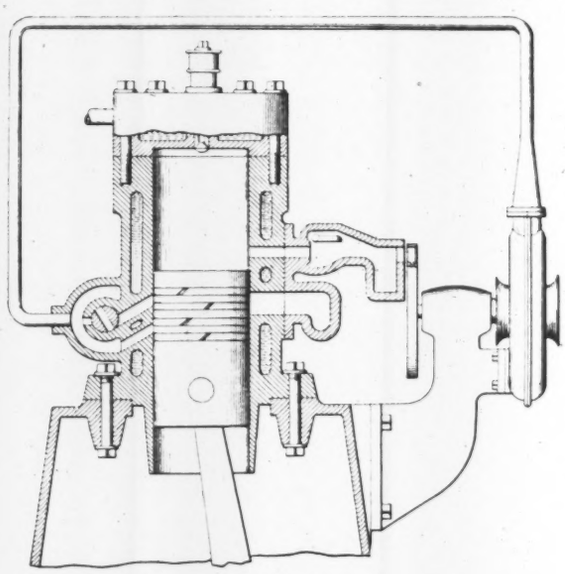

Fig. 10 Scavenging a Two-Cycle Diesel Engine with an Exhaust-Gas Turbine

in. and appreciable gains in power and savings in fuel consumption obtained.

A few unpublished experiments have been made with an exhaust-gas turbine supercharging a two-cycle engine according to a plan devised by the author and shown in Fig. 8; this is explained later in the paper.

Theory of Supercharging a Diesel Engine

As already explained, supercharging involves forcing air into the cylinder by a compressor or blower of some kind, so that at the beginning of compression there exists within the cylinder an absolute pressure greater than atmospheric. Without supercharging, the pressure would be below atmospheric for four-cycle engines. In the case of two-cycle engines, supercharging means increase of the pressure of scavenging air so that the absolute pressure in the cylinder at the beginning of compression is greater than the value slightly above atmosphere which ordinarily serves to eject the exhaust gases. Of course, the two-cycle engine can only be supercharged if there is some valve or port arrangement which permits of adding additional charge after the exhaust ports are closed.

Such supercharging changes the power delivered by the Diesel engine in the following ways which are later considered in detail.

In the first place, if the inlet and exhaust valves have a proper lap, the supercharging pressure can result in scavenging the clearance volume of exhaust gases so that the air charge is greater than the displacement due to the filling of the clearance volume with air at the supercharging pressure.

In the second place, the removal of the exhaust gases gives a cleaner charge, particularly in the vicinity of the fuel injection nozzle, which has been found to permit of the burning of a greater amount of oil—that is, a decreased air excess.

In the third place, the increase of power with a given water-jacketed surface can decrease the percentage of heat carried away by the cooling water, which increases both power and thermal efficiency.

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[1] Von Hellmann, *Zeitschrift des Vereines deutscher Ingenieure*, July 29, 1922, no. 30, p. 1281, continued to Aug. 5, 1911, no. 31, with a bibliography. Also Jan. 17, 1914, no. 3, p. 105.

Unpublished experiments made by the Allgemeine Elektricitäts Gesellschaft on a 1500-hp. Burmeister & Wain engine in 1922–3. Later experiments by the Allgemeine Elektricitäts Gesellschaft are included in an article by Dr. W. Scholz in *Werft, Reederei, Hafen*, June 22, 1924, p. 269, and in *Motorship* (London), Sept., 1924, p. 207, and Nov., 1924, p. 294.

A noteworthy set of experiments made by Dr. W. Riehm is described in *Zeitschrift des Vereines deutscher Ingenieure*, Aug. 4, 1923, No. 31 p. 763. This article is a paper read by Riehm at the Diesel Engine Congress, Vereines deutscher Ingenieure, June 29, 1923, the papers of which have been published complete in English. It is also abstracted in *Motorship* (London), Nov., 1924, p. 294.

In the United States, the Falk Company, in Milwaukee, published some experiments with supercharger pressures in the neighborhood of 1 lb. per sq. in., see *Motorship* (N. Y.), Oct., 1924, p. 734.

In the fourth place, the production of an increased amount of power with given cylinder and crank friction increases the mechanical efficiency.

In the fifth place, the expansion of an increased charge to a given final volume gives a decrease in the area of the theoretical indicator card.

The relative effect of each of these items, of course, varies with different engines.

Clearance and Compression. The amounts of supercharging ordinarily contemplated, say, from 1 lb. per sq. in. to 5 lb. per sq. in. above the absolute pressure of the atmosphere, involve an increase of the amount of charge of from 6 to 30 per cent. If the clearance volume with supercharging is maintained at the same percentage as without, there will result an increase of compression pressure, and this has occurred in some of the cases previously mentioned. In most cases, however, the clearance volume has been increased in the same ratio as the absolute inlet pressure, so as to maintain the original compression pressure.

Scavenged Clearance Volume. Whether there is supercharging or not, the inlet valve always opens just before the piston reaches the end of the exhaust stroke, in order to properly start the admission of the new charge. Whether there is supercharging or not, the exhaust valve always closes somewhat after the end of the exhaust stroke and when the piston has started on the suction stroke. This gives a period when the piston is near the top of the stroke at the end of the exhaust and the beginning of the admission, when both inlet and exhaust valves are open together. When a supercharger is used, there is a pressure in the inlet manifold appreciably higher than that in the exhaust manifold, and hence the new charge will start to enter the cylinder even during the latter part of the exhaust stroke. Therefore some of the exhaust gases in the clearance space, which ordinarily are never pushed out by piston displacement, will be forced out by the supercharging pressure.

If the valve openings are properly retimed to suit supercharging, it is quite possible that the clearance space may be filled with fresh charge and the exhaust gases entirely removed without discharge of air through the exhaust port. As the piston proceeds on the suction stroke with the exhaust valve fully closed and the inlet valve fully open, the cylinder (including the clearance space) will then be filled with a fresh charge at a pressure below the pressure of the supercharger by the amount of the valve and port friction. The increase of charge due to supercharging is due to filling the cylinder at this increased pressure as well as to replacing the exhaust gas in the clearance space.

If the original compression pressure is maintained, the clearance volume must be increased in the ratio of the absolute pressures at the beginning of the compression stroke with and without supercharging. Hence the clearance space which is filled with the added fresh charge in the case of supercharging is larger even than the clearance space in the original engine.

Curves are given in Fig. 11 showing the percentage increase in amount of new charge due to all of these facts, with pressures of $^1/_2$, 1, and 2 lb. per sq. in. below atmospheric pressure in the cylinder at the beginning of the compression stroke and various amounts of supercharging.

The increase in power with given supercharging depends not only on the theoretical increase in charge as given by the curve, but also on the amount of fuel burned with a given amount of charge, which depends upon considerations given beyond.

Improvement of Combustion Due to Supercharging. Some of the experiments previously cited have given an increase of power with Diesel engines greater than that theoretically computed by the principles of the preceding paragraphs, due to the fact that it has been found possible to burn greater percentages of oil. As is well known, the charge of oil injected into the air of a Diesel engine requires about half of the air for theoretical combustion. The amount of oil injected at maximum power is always made as great as is consistent with a clean exhaust. The absence of exhaust gases from the clearance volume facilitates the chemical act of combustion and the rapidity with which it spreads. The increased pressure with which the supercharged air enters the cylinder through the inlet valve also creates currents so that there is certain to be pure air in the vicinity of the fuel-injection nozzle. Without supercharging there is, of course, a pressure in the exhaust manifold higher than in the inlet manifold, so that during the valve lap previously referred to, exhaust gases may actually enter the inlet manifold and be returned again into the cylinder near the fuel-injection nozzle when the exhaust valve closes. All of these actions explain the well-verified fact that an increased percentage of fuel can be burned in a Diesel engine when supercharging is used, giving a decreased air excess. Hence there has been the great increase in engine power obtained with a small amount of supercharging, 400 to 600 millimeters of water, used in many recent German installations.

Water-Jacket Loss. If in an unsupercharged and a supercharged Diesel engine the same compression pressure and the same percentage of air charge are required for the combustion of the oil, there will be an increased amount of heat liberated with supercharging without any change in temperature. The surface of the combustion chamber in the two cases will be nearly the same, and hence there will be the same amount of heat lost to the jacket water during the early part of the stroke. The jacket-water loss will then be a smaller fraction of the total heat of combustion, which will result in a slight improvement in efficiency. If, however,

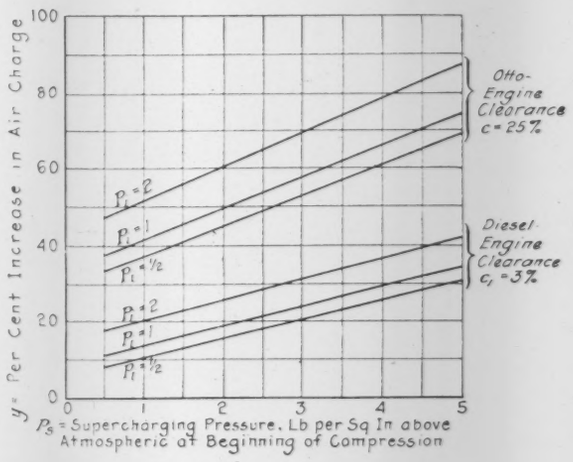

FIG. 11 THEORETICAL INCREASE IN AIR CHARGE DUE TO SUPERCHARGING (P_1 = initial pressure without supercharging in cylinder at beginning of compression, lb. per sq. in. below atmospheric.)

the percentage of air which is used to burn the fuel is increased and the air excess diminished, there will of course be an increase of temperature and an increase of the jacket-water loss. However, the considerations previously mentioned will tend to keep this a small percentage for the supercharged case. It is true that the combustion in a supercharged engine will last through a somewhat greater portion of the stroke than otherwise, and the temperature when the exhaust valve opens will be somewhat higher. This will tend to increase the jacket-water loss during the latter part of the stroke where it is comparatively small. On the whole, however, it is to be expected that supercharging will decrease the relative amount of heat lost to the jacket.

Increase in Mechanical Efficiency. The increase of power from a given cylinder due both to the increased fuel ratio and the increased charge, decreases the relative percentage of friction loss. This is a major item and is the primary reason for the observed increase in thermal efficiency which has always accompanied supercharging. The theoretical loss due to decreased ratio of expansion mentioned in the next paragraph diminishes this gain somewhat, and the decreased jacket-water loss mentioned above increases it somewhat, but both of these are probably minor items.

Incomplete Expansion. In an engine without supercharging the volume of the gas at the end of expansion is equal to the volume occupied by the initial air charge when at a pressure slightly below atmospheric pressure, both of these volumes being equal to the cylinder displacement. With a supercharged engine the volume of the gases at the end of expansion is equal to the cylinder displacement as above, but this volume is now equal to the volume of the initial air charge when at a pressure appreciably above atmospheric pressure. Hence a supercharged engine does not expand the gases to as low a pressure as in the normal case. In other words,

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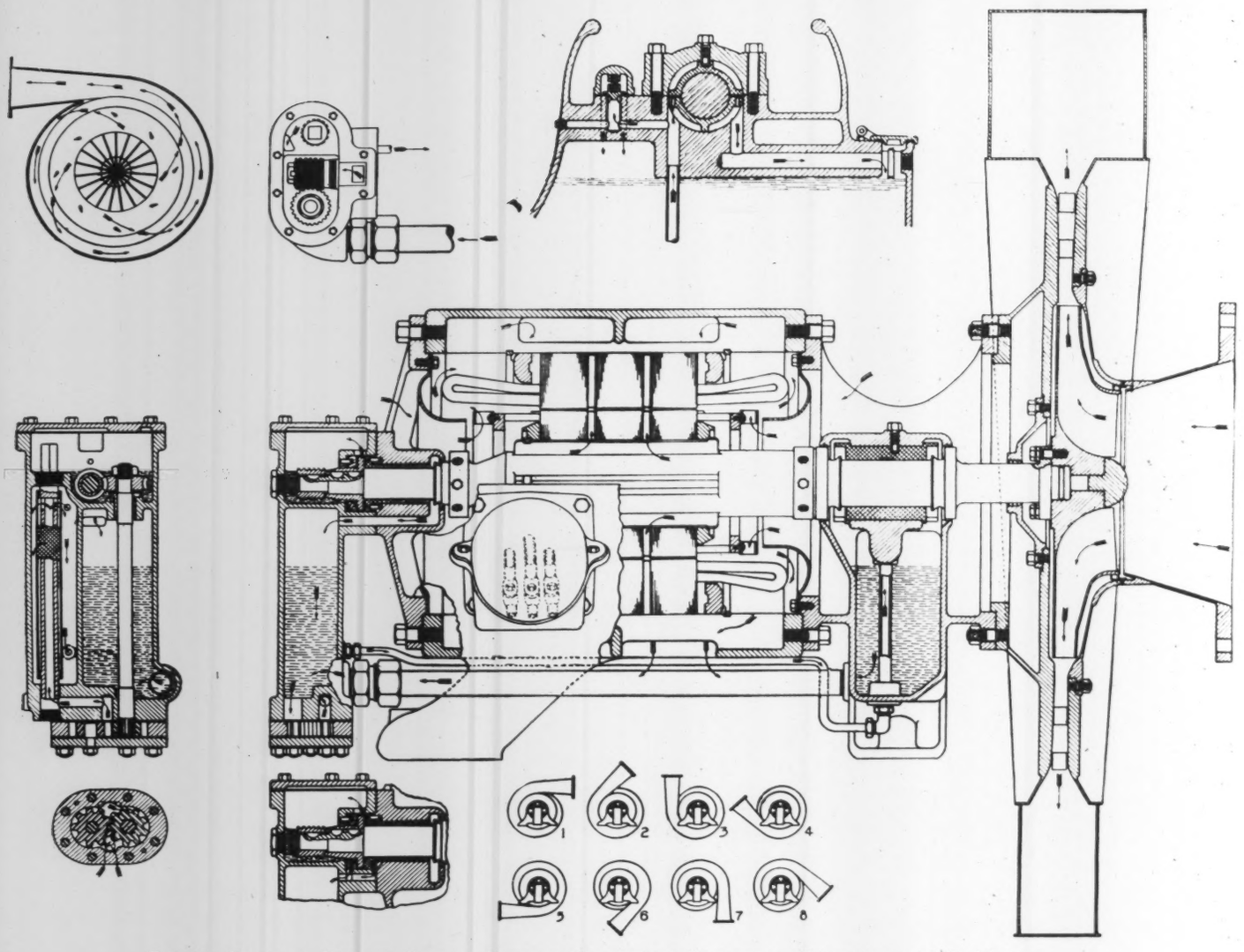

Fig. 12 Sectional View of General Electric Centrifugal Compressor Driven by an A.C. Motor

there is a small area at the toe of the theoretical indicator card which is lost. The indicated pressure in the cylinder is comparatively small at this end of the stroke and is but little greater or perhaps not as great as the pressure necessary to overcome the engine friction. Hence the loss in an actual engine is little or nothing.

Proposals have often been made with both Diesel- and Otto-cycle engines to obtain a theoretical gain by extending the expansion and increasing the length of the toe of the card beyond that in a normal engine so that the volume at the end of expansion would be greater than the volume of the initial air charge when at atmospheric pressure. This would replace the theoretical loss which occurs with supercharging by a theoretical gain. The effect of engine friction would undoubtedly reverse the situation, however.

Temperatures with Supercharging. It must be borne in mind that supercharging of itself with a given air excess involves little or no increase of temperature. In fact, in some of Riehm's experiments with a very large amount of supercharging there was an increase in power with a decreased percentage of air required for the combustion of the oil—that is, an increased air excess. This resulted in decreased exhaust temperatures and a decreased jacket-water loss.

If, however, advantage is taken of the fact that the use of supercharging also permits of a decrease in the air excess, increased temperatures will of course result. This gives stress and cooling problems which must be properly faced.

If a Diesel engine of a given design is barely able to deliver its maximum power with no trouble with valves, cylinder heads, or other parts, it would of course be futile to attempt supercharging of this kind. However, the fact that a number of engines are successfully operating with supercharging is prima facie evidence that Diesel engines can be designed to withstand the increase in the amount of power for a given cylinder displacement. There is no doubt that there are types of engines now in use which would give trouble if supercharging were attempted, but the success of other types with superchargers will lead to such alterations as will permit of a successful solution.

During the work done on the supercharging of airplane engines in the United States the supercharging has been pushed beyond the point of obtaining normal power at sea level and some supercharging has been done at low altitude, which has been called "supercharging below sea level." This, for an Otto-cycle engine, is the same thing proposed here for a Diesel engine, and the use of special steel alloys has permitted the successful operation of valves in these airplane engines at temperatures much greater than can ever be encountered in a Diesel engine.

In fact, it may be safely said that it is eminently practical to supercharge with a pressure of a few pounds per square inch, with an increase of clearance volume so as not to increase the compression pressure, and with but slight diminution in the air excess so as to avoid excessive exhaust temperatures. It is to be hoped, however, that future researches will permit of supercharging with pressure of 5 lb. per sq. in. or more, without increase of clearance volume but with increase of compression pressure and with increase of oil supply to the maximum amount which will give a clean exhaust. This will incorporate in one engine all of the advantages which have been gained independently, and will give a truly astonishing gain in thermal efficiency and power for a given engine weight.

Supercharging Two-Cycle Diesel Engines. Some manufacturers of Diesel engines have such valve arrangements as will give opportunity for entrance of scavenging air after the exhaust port is closed

by the piston on the up stroke, and this has been called supercharging. However, this arrangement or any other arrangement heretofore used has little more than atmospheric pressure in the cylinder when the port was finally closed and compression begun. This is shown by the fact that two-cycle engines with the most effective scavenging give a mean effective pressure for the whole stroke of about 122 lb. per sq. in. The lower end of the stroke is of course lost in a two-cycle engine, so that the effective stroke is about 77 per cent. This gives an equivalent mean effective pressure of 158 lb. per sq. in. during the effective stroke, which corresponds to values obtained by a good four-cycle Diesel engine. This means that the best two-cycle engine has a charge at the beginning of compression equal to that of a four-cycle engine. There must be a greater charge than this to give actual supercharging. This would require greater pressures and volumes than have ever been provided with reciprocating blowers, but these could easily be provided with centrifugal compressors. If the cylinder head, piston, and cooling arrangements can be constructed to stand the increased power, the centrifugal compressor can be arranged so that the air pressure in the cylinder at the beginning of the compression stroke will be above atmospheric pressure. This will result in the same increase of power due to supercharging which is discussed elsewhere for four-cycle engines. The higher pressure could be used for the scavenging also, which would then be done more expeditiously, so as to make a greater percentage of the stroke effective. It would also be possible to provide two centrifugal compressors, one such as is now used for scavenging, and a second one providing air at a higher pressure through a mechanical valve opened at the proper time.

EXHAUST-GAS TURBINES FOR SCAVENGING AND SUPERCHARGING

In 1911 Dr. Wilhelm Schmidt proposed a centrifugal compressor driven by an exhaust-gas turbine for supercharging a four-cycle Diesel engine,[1] and the proposal was repeated by Reihm. The same plan has since been extensively and successively used by Rateau in France and by the General Electric Company in the United States for airplane engines.

With this system the exhaust gases of a Diesel engine, instead of passing to the atmosphere, pass to an exhaust manifold and a nozzle box wherein is maintained a pressure above atmosphere, giving a back pressure against which the engine exhausts. The expansion of the gases through the nozzles from this back pressure to atmospheric pressure gives power which drives a turbine wheel. This turbine drives a centrifugal compressor which compresses air to a pressure above atmosphere and delivers it to the intake manifold. The engine displacement would then be filled with a charge at a pressure above atmosphere, just as in the case of a supercharger driven by a separate electric motor. In such a case the power for supercharging would be obtained by the decrease of the mean effective pressure shown by the indicator card, due to the increased exhaust back pressure. The theoretical efficiency is the same with a separately driven supercharger or an exhaust-gas-driven supercharger. The actual efficiency would depend upon the relative efficiency of transfer of power from the cylinder mean effective pressure, through the crankshaft, electric generator, and electric motor in the one case, and the efficiency of the turbine wheel in the other case. These two efficiencies are probably nearly equal. Data do not exist for a complete computation in the matter. Fig. 9 shows an actual apparatus of this kind advertised by the Brown-Boveri Company.

In the airplane installation the engine exhaust pressure is nearly equal to the supercharger pressure, and the difference in temperature enables the gas turbine to drive the compressor without mechanical connection to the engine. It is to be expected that a similar situation would exist with a Diesel engine. The scavenging of the clearance space would be lost with this plan.

Exhaust-Gas Turbines with Extended Expansion. Modugno[2] has put forward a number of other plans of using an exhaust-gas turbine with a Diesel engine. He proposes to obtain the "extended expansion" and theoretical gain due to increasing the length of the toe of the card, without supercharging, by gearing to the engine shaft an exhaust-gas turbine driven by increased engine-exhaust back pressure. He also proposes to obtain extended expansion as well as supercharging by using an exhaust-gas turbine with high back pressure which delivers considerably more power than necessary to drive a centrifugal supercharger.

The exhaust back pressure would then be greater than the supercharger pressure, so that these plans differ from Schmidt's exhaust-gas turbine by delivering power to the turbine wheel by raising the back pressure, which must be utilized by gearing. The author feels that in both of these plans the losses in the turbine wheel and gearing would be more than the theoretical gain due to the extended expansion.

Exhaust-Gas Turbine for Scavenging a Two-Cycle Diesel Engine. Fig. 8 shows a plan devised by the author which has been made the subject of a few tests and an extended series of computations. An extra exhaust port is provided at the side of the cylinder somewhat in advance of the regular exhaust port. In this port is a check valve opening outward into a header which leads to the nozzle box of an exhaust-gas turbine. Four to six cylinders of a two-cycle engine must be connected to the header. When any one of the pistons passes the above-mentioned extra exhaust port during the expansion stroke, there will be sufficient pressure to open the check valve and pass a certain amount of exhaust gas to the nozzle box. This will continue until the piston begins to open the regular exhaust port. The pressure in the cylinder will then drop to a value below that in the nozzle box and the check valve for this particular cylinder will close. The exhaust will then pass out through the regular exhaust port and air will enter through the regular scavenging port just as in an ordinary two-cycle engine.

During the upward or compression stroke the pressure in the cylinder will be less than at the corresponding point in the down or expansion stroke, and the pressure in the nozzle box from other cylinders will prevent the check valve from opening. Hence, from the time that the check valve closes on the down stroke, the events in the cylinder will be exactly the same as for any two-cycle engine.

Thus, as each piston passes the extra exhaust port a charge of exhaust gas will pass into the nozzle box. With a six-cylinder engine there will be six such charges admitted during each revolution, and this will maintain a somewhat fluctuating pressure in the nozzle box. The magnitude of this pressure will, of course, depend upon the point in the stroke at which the extra exhaust port is placed. This point can be made anything which is found necessary in order to produce such pressure as will give the necessary amount of power for scavenging. As stated, the pressure in the nozzle box will fluctuate somewhat due to the successive charges admitted. This pressure is bound to be low enough during the time when each piston passes its extra exhaust port to permit the check valve to open, and the nozzle-box pressure will also be high enough to close the check valve as soon as the regular exhaust port is opened and to keep it closed during the rest of the down stroke and the early part of the compression or upstroke.

There will be provided a nozzle box, turbine wheel, and centrifugal compressor similar to those of an airplane supercharger. The proportions will be different, but the general design, materials, and methods of construction will be the same.

A certain amount of power is required to operate the scavenger for any two-cycle Diesel engine, and this power of course must be subtracted from the gross power which would otherwise be delivered from the engine. In the case of the present method this power is taken from the indicated horsepower. At the time the piston passes the extra exhaust port there will be an appreciable pressure in the cylinder, and the opening of this exhaust port will of course reduce this pressure below the value which would otherwise have existed during the remainder of the expansion stroke. An engine with an equipment of this kind will therefore have a jog in the expansion line near the end of the stroke which will make this line somewhat lower than it otherwise would have been down to the beginning of the exhaust. There will be an area between the expansion line as so lowered and the ordinary expansion line which will represent the power required for scavenging. This power will not pass through the piston rod and crankshaft as in the case of a direct-driven reciprocating scavenger, nor will it have to be produced by a separate generator and motor, with consequent

[1] German patents Nos. 281394 and 286050, *Zeitschrift des Vereines deutscher Ingenieure*, Nov., 25, 1916, p. 982.

[2] Trans. Inst. Naval Architects, England, 1923, vol. 65, p. 321.

losses as in the case of a separate centrifugal compressor. There will be instead losses due to some pressure drop through the check valve and the losses of the turbine wheel. Some computations made on this matter indicate that all of these different kinds of losses practically counterbalance each other, and that this method of scavenging will produce about the same amount of net power from Diesel engines as any other method. The proposed method is expected to give a great saving of weight and space, due to the fact that the apparatus consists merely of a turbine wheel and centrifugal compressor running at 10,000 to 20,000 r.p.m.

A great deal of the material of this paper has been obtained from the files of various periodicals and an effort has been made to give complete references in all cases. The author also wishes to acknowledge assistance from Messrs. W. W. Eisenwinter, W. E. Ver Planck, M. G. Robinson, B. L. Spain and W. G. Mulheron, of the General Electric Company.

Synopses of Other Annual Meeting Papers

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Production Control in the Newsprint Industry

By GEORGE D. BEARCE

Engineer, Newsprint Service Bureau, New York, N. Y. Assoc-Mem. A.S.M.E.

THIS paper outlines the manufacturing processes and production methods of a majority of the mills in North America making "standard" newsprint paper. The general production controls used in the "groundwood," "sulphite," and "paper" departments are described and illustrated by representative forms.

The manufacturers have adopted standard methods of calculating three "operating reports" and four "conversion cost reports" covering important operations in the pulp and paper departments of the plant. The three "operating reports" are described in detail and illustrated by forms.

Reports are exchanged on a reciprocal basis and methods of compilation and distribution are discussed and illustrated. The purpose is to supply the participating companies with current information that will be helpful to them in the installation of operating economies.

Safety in Materials Handling

By DAVID S. BEYER

Vice-President and Chief Engineer, Liberty Mutual Insurance Company, Boston, Mass.

AFTER demonstrating the seriousness of the materials-handling accident hazard by showing that accidents attributed to materials handling contribute approximately one-third of all industrial accidents, the author computes the yearly economic loss to be $250,000,000. He then analyzes briefly such accident hazards in general broad classifications and points out a few methods of prevention. He calls the attention of engineers to the necessity for including in construction drawings and specifications means for the prevention of such accidents.

Radiation in Boiler Furnaces

By B. N. BROIDO

Consulting Engineer, The Superheater Company, New York, N. Y. Mem. A.S.M.E.

THERE is now a noticeable tendency toward an extensive application of water-cooled furnace walls for absorbing heat by radiation. An analysis of the fundamentals of radiation and of the effect of water-cooled walls on the gas temperatures in the furnace is given in this paper.

There is a certain relation between the amount of fuel burned, the surface exposed to direct radiation in the furnace, and the total heat absorbed by this surface. Curves are given showing this relation for different installations and a standard curve is drawn which enables the designer to determine with sufficient accuracy for practical purposes what part of the total heat generated in the furnace at different ratings is absorbed by the water-cooled walls. This curve can also be used to find the heat transmission by radiation per square foot of heating surface for any given condition of the furnace.

The amount of radiant-heat-absorbing surface that can be installed advantageously in a furnace depends upon the type of furnace and the kind of fuel burned. Based on a number of installations with water-cooled walls, the most advantageous arrangement of such surface for given conditions is suggested.

The effect of radiation from gases on heat transmission—particularly with high temperature differences—and the influence of radiation upon the measurement of gas temperatures are briefly discussed.

Phosphor-Bronze Helical Springs from the Standpoint of Precision Instruments

By W. G. BROMBACHER

Physicist, Bureau of Standards, Washington, D. C.

THIS paper gives the results of tests made on phosphor-bronze helical springs investigated at the Bureau of Standards to obtain knowledge useful in the design of springs for precision instruments. The characteristics of the spring material, the method of construction of the springs, the apparatus in which the springs were tested, and the procedure followed are set forth by the author. The results relate to stiffness, maximum fiber stress, hysteresis, after-effect, drift, and buckling.

Tangential Vibration of Steam-Turbine Buckets

By WILFRED CAMPBELL AND W. C. HECKMAN

Wilfred Campbell died at Schenectady, N. Y., July 7, 1924. Before his death he had intended to present this paper which is composed of letters and memoranda in his handwriting or dictated by him. Few alterations have been made other than those required to fit the fragments into a connected story.

AT THE Spring Meeting of this Society in 1924 a paper was presented by the author on The Protection of Steam-Turbine Disk Wheels from Axial Vibration. The present paper describes how the same research was extended to include tangential vibration. Substantially the same testing apparatus was used and the same methods of protection have been adopted. Resonant conditions are avoided by carefully prescribed margins. In unsymmetrical reaction buckets the component vibration lying most nearly in the plane of the wheel is, by definition, treated as a tangential vibration. The methods described suffice for protection against any combined or intermediate type. Mr. Heckman, of the Turbine Engineering Department of the General Electric Co., contributes additional results in Part II of the paper.

Recent Developments at Colfax Station, Duquesne Light Co.

By CHARLES W. E. CLARKE

Dwight P. Robinson & Co., New York, N. Y. Mem. A.S.M.E.

THIS is the fourth of a series of papers dealing with the Colfax Station of the Duquesne Light Co., Pittsburgh, Pa., which the author has presented to the Society. In the present paper are given the test results of the 3-A element of the station which showed net heat rates of 12,750 B.t.u. per kw-hr. at 30,333 kw., 13,021 B.t.u. per kw-hr. at 22,400 kw., and 14,200 B.t.u. per kw-hr. at 15,050 kw. A brief history of the physical plant involved in the original station, and in the unit extensions Nos. 1, 2, and 3, is given, with data on boiler and turbine outages and condenser cleaning. The author shows that the total cost of the station has been to date $115.80 per kw. for the 190,000 kw. of installed capacity, and predicts a total cost, when the fourth unit is installed, of $107.50 per kw. for 270,000 kw.

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Normal Pitch—The Index of Gear Performance

By G. M. EATON

Chief Mechanical Engineer, Westinghouse Electric & Manufacturing Co., East Pittsburgh, Pa. Mem. A.S.M.E.

THIS paper brings out certain departures from previously accepted practice which are useful in the manufacture of heavy involute gearing, as they ease the performance during the breaking-in stage of operation. It shows that material improvement in performance may be secured by adopting the proper relation between the normal pitches of the driving and driven gears, measured at the point of tooth engagement. Finally, it outlines the development of normal-pitch indicators.

Formulas for the Design of Helical Springs of Square or Rectangular Steel

By C. T. EDGERTON

Bureau of Statistics, New York, N. Y.

ALTHOUGH helical springs of square- or rectangular-bar steel are not common, problems involving certain extreme requirements are occasionally encountered which can be met only by such types of springs. The author points out the lack of formulas for calculating any except springs of square-bar steel, and then develops formulas for rectangular-bar steel based on the work of St. Venant. For the solution of these formulas he gives tabulated values for two variables which depend on the ratio of the bar's cross-sectional dimensions. An appendix contains four examples in which the application of the formula is illustrated.

A Graphical Study of Journal Lubrication (Part III)

By H. A. S. HOWARTH

General Manager, Chief Engineer, Kingsbury Machine Works.

THIS paper continues the investigation of journal lubrication reported to the Society under the same title in 1923 and 1924. Friction curves are here presented for central and offset partial bearings whose curvature radius exceeds that of the journal. The characteristics of fitted partial bearings are then studied, including their carrying capacities and friction.

Torsional-Stress Concentrations in Shafts of Circular Cross-Section and Variable Diameter

By LYDIK S. JACOBSEN

Formerly in Engineering Department of Westinghouse Electric & Manufacturing Company, East Pittsburgh, Pa.; now instructor in Mechanical Engineering, Stanford University, California.

THE object of this paper is to point out an electrical, experimental method for finding the torsional-stress distribution in cylindrical shafts of circular cross-section and of any axial outline. The method has been applied to the problem considered by F. A. Willers, and the paper includes a series of curves that will enable the designer to find, at a glance, the maximum torsional stress in a shaft of two diameters for various diameter and fillet proportions. It should be remarked that the results for a shaft of two diameters may be applied to most shafts of several diameters, since the length of any two adjacent portions of the shaft are usually sufficiently long to make the analysis apply. Special shaft outlines, for instance shafts with oil throwers, can be investigated by the method.

An Outline for the Application of Fatigue and Elastic Results to Metal-Spring Design

By T. McLEAN JASPER

Special Research Associate-Professor of Engineering Materials, University of Illinois, Urbana, Ill.

THIS paper deals particularly with steel springs used for shock-absorbing purposes and for recuperating machinery. The problem of the design of springs may be divided into two parts: first, the question of the static elastic and fatigue properties of the material to be used in their construction, and second, the shape of the springs desired, together with the distribution of the stresses developed in their use for a given deformation. The paper includes some results of an investigation of static properties of steel carried out at the Engineering Experiment Station of the University of Illinois.

Charts for Studying the Oil Film in Bearings

By GEORGE B. KARELITZ

Westinghouse Research Laboratory, East Pittsburgh, Pa.

IT HAS been shown several times that the hydrodynamical theory of lubrication satisfactorily explains the mechanism of lubrication. Since the necessary calculations are very complicated, the practical application of the theory, up to the present time, has been very difficult. The charts presented in the paper give the designer or investigator a means of determining with sufficient accuracy the shape and pressures in the oil film for bearings under different conditions.

Vibration Phenomena of a Loaded Unbalanced Shaft while Passing through Its Critical Speed

By A. L. KIMBALL, JR. and E. H. HULL

Both of Research Laboratory of General Electric Co., Schenectady, N. Y., and Respectively Associate and Junior Members A.S.M.E.

THE purpose of this paper is to present in a clear manner, with experimental demonstration, a discussion of the peculiar vibration phenomena to which an unbalanced loaded shaft is subject while passing through its critical speed. First the basic theory is given from a somewhat new viewpoint, after which an experimental demonstration, which gives a quantitative check on the theory, is described. As far as the authors are aware, no such demonstration has thus far been made, except in a qualitative way.

Some Comparative Wear Experiments on Cast-Iron Gear Teeth

By GUIDO H. MARX, LAWRENCE E. CUTTER, and BOYNTON M. GREEN

Respectively Professor of Machine Design, Stanford University, Mem. A.S.M.E.; Associate Professor of Mechanical Engineering, Stanford University, Mem. A.S.M.E.; and Assistant Professor of Mechanical Engineering, Stanford University Assoc-Mem. A.S.M.E.

THE experiments reported in this paper had their origin in a suggestion of Mr. Luther D. Burlingame, and were made in the laboratories of the mechanical engineering department at Stanford University. The deductions indicated by the tests made are as follows:

(*a*) The standard-depth 20-deg. involute tooth form appears to be a better one to resist wear than the standard-depth $14^1/_2$-deg. involute form.

(*b*) The stub-tooth, 20-deg. involute tooth form appears to be a better one to resist wear than the standard-depth $14^1/_2$-deg. involute form.

(*c*) The standard-depth 20-deg. involute tooth form appears to be a better one to resist wear than the stub-tooth 20-deg. involute form.

(*d*) One series of tests indicates predominance of the effect of pressure angle over the effect of ratio of arc of action to pitch arc.

(*e*) Two other series indicate the importance of the effect of ratio of arc of action to pitch arc in distributing the load over several teeth, when employing the same pressure angle; and indicate that this effect more than counterbalances disadvantages due to the longer paths of relative rubbing and higher frictional velocity.

Auxiliaries for Motor Vessels

By JOHN W. MORTON and A. B. NEWELL

Respectively, Engineer in Charge, Diesel Department, N. Y. Shipbuilding Corp., Camden, N. J. Mem. A.S.M.E., and Guarantee Engineer, N. Y. Shipbuilding Corp., Camden, N. J.

WITH the vast increase in motorships it is only natural to expect a variety of installations; and the many designs and variety of types of propelling machinery and modes of drive in turn affect the auxiliary equipment.

The great differences that exist in auxiliaries in such factors as weight, space, duty, capacity, cost, etc., without doubt could be minimized.

Determining the right kind and capacity of auxiliaries, in order to reduce first cost and weight and at the same time have the most economical marine plant both at sea and in port, is a somewhat different task, because one has to consider the ship's trade routes, and whether the ship is new or converted.

It must also be remembered that the assumption regarding continual running with fuel cargo is rather ideal, but as all ships will be equally liable to runs in ballast or with part cargo or load, the results will be relative.

Comparisons between types of machinery are often based on the relative space required, the weight and fuel-consumption values, and the resulting saving in fuel.

While perhaps these are quite fair to the machinery, they are not to the ship as a whole, and are apt to show differences, the benefits of which in practice are never realized.

The comparisons in the paper have been made without bias for or against any particular type of machinery, and in a manner that shows a truer ultimate saving than that attained by other comparative methods.

The Tension Ratio and Transmissive Power of Belts

By C. A. NORMAN

Professor of Machine Design, The Ohio State University, Columbus, Ohio. Mem. A.S.M.E.

AFTER reviewing the tests of Lewis, Kammerer, and Friedrich, and those at Cornell University and the Mellon Institute, the author shows that an investigation of the increase of transmissive power with slip was warranted. Such an investigation on an apparatus patterned after that of Friedrich was conducted at The Ohio State University on rubber, leather, and fabric belts. The results of the investigation are given in the form of curves, and the conclusions drawn from the tests are noted.

The Heat-Balance Method of Testing Centrifugal Compressors

By M. G. ROBINSON

General Electric Co., River Works, Lynn, Mass. Assoc-Mem. A.S.M.E.

CENTRIFUGAL compressors are often driven by steam turbines with and without intervening gear trains. To determine the shaft power transmitted to the compressor by its driver from efficiency tests on the turbine and estimates of the gear losses, involves considerable labor, time, and expense. The heat-balance method described by the author is simpler and more direct. In this method the principle is used that a balance exists between the mechanical energy supplied by the rotation of the shaft and the heat-energy increase in the air which is being compressed, the heat-energy increase in the cooling water used, and the heat lost from the casing, bearings, and packings. With proper testing apparatus and due precautions this heat energy may be accurately measured, and hence the power input established. Very satisfactory results have been secured by the application of this method. There are given details of the fundamental theory, some representative test data, and a complete description of the apparatus used.

The Value of Higher Steam Pressures in the Industrial Plant

By WILLIAM F. RYAN

Power Engineer, The Solvay Process Co., Syracuse, N. Y. Mem. A.S.M.E.

THIS paper aims to point out the value of higher steam pressures in the industrial plant. It is shown that the gains from high pressure are greater than in the central station, and except for the question of suitable feedwater for high-pressure boilers, the problems involved are less difficult. The relative cost of power for varying initial pressure is estimated. The relative efficiency of turbine and engine prime movers, and the application of higher pressures to manufacturing equipment, are discussed. It is indicated that pressures up to the present commercial limit, about 1200 lb. per sq. in., may be used advantageously in the industrial plant which has a high load factor, and in which the demand for power is high, in proportion to the demand for process steam.

Heat-Treatment Data on Quality Steel Castings

By ALBERT E. WHITE

Professor of Metallurgical Engineering, University of Michigan, Ann Arbor, Mich. Mem. A.S.M.E.

THIS paper summarizes a study which has been made on heat-treatment practice for quality steel castings. It gives the results of laboratory tests on dendritic and dendrite-free steels after an annealing, a normalizing, a drawing, and a spheroidizing treatment. It likewise records a large number of plant tests. The results show that a normalizing and drawing treatment gives superior results to those obtainable by annealing or spheroidizing.

Steam Bleeding and Turbine Performance

By C. D. ZIMMERMAN

Production Engineer, Steam Department, Cleveland Electric Illuminating Company, Cleveland, Ohio. Mem. A.S.M.E.

THE primary object of this paper is to give the results of tests which show the improvement in overall steam power-plant economy due to single-stage steam bleeding. Detailed results of the tests are also described which show the changes due to steam bleeding in boiler-unit efficiency, degree of superheat, and the performance of the turbine-condenser unit.

In order to complete the study of the effects of steam bleeding it was necessary to show the effects of changes in superheat and condenser back pressure on the turbine performance. The results of these tests are given in the second part of the paper and are of interest as a separate study of steam-turbine performance. With this in mind, additional data such as the effect of air in the turbine exhaust steam and effect of changes in initial steam pressure are also given. The change in superheat covered the range from 100 to 300 deg. fahr. and the condenser back pressure was of sufficiently low value as to show the economical value for loads above 25,000 kw.

The tests were made at the Lake Shore Station of the Cleveland Electric Illuminating Co. and the turbine tested was a Curtis-type, 17-stage machine of 30,000 kw. capacity. Steam was bled from the 13th stage.

Contributors to this Issue

V. J. Azbe

Victor J. Azbe, author of the paper on Industrial-Furnace Efficiency, was born in 1889 in Laibach, Jugo-Slavia. He has had nine years' experience in the operation of boiler plants with Swift & Co. and the Anheuser-Busch Brewing Co., where he supervised the testing. He has been engaged in consulting work in St. Louis, Mo., since 1918. He also conducts a testing and research laboratory.

Mr. Azbe has contributed a number of papers to technical publications. He is active in the affairs of the St. Louis Section of The American Society of Mechanical Engineers.

* * * * *

H. H. Blee

Harry H. Blee, author of the paper on The Airship and Its Place in Commerce, received his A.B. degree in mechanical engineering from Leland Stanford, Jr. University in 1910. He remained to carry out post-graduate work and the following year received his M.E. degree, after which he spent still another year there as instructor in mechanical engineering. In 1912 he became mechanical engineer for R. F. Chevalier, industrial and power-plant consultant, of San Francisco. From 1913 to 1917 he was engaged in mechanical and civil engineering work with the California State Department of Engineering.

Upon the outbreak of the war Mr. Blee entered the Airplane Engineering Division, U. S. Army Air Service, and was soon commissioned captain and appointed chief of the Publications and Technical Data Department, with which he served until 1919. After leaving the service Mr. Blee rejoined the California State Department of Engineering as water-supply engineer. He resigned in 1921 to establish a private engineering practice in Southern California. For the past year Mr. Blee has been a student officer at the Air Service Balloon and Airship School, Scott Field, Ill. He is a major in the Air Service Reserve and has received the military ratings of balloon observer and airship pilot, and the Fédération Aéronautique Internationale ratings of spherical balloon pilot and airship pilot.

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C. M. Bonnell, Jr., author of the paper on Scientifically Developed Shipping Containers, is a graduate of the Stevens Institute of Technology, where he received his degree of M.E. in 1919. For a short time after graduation he remained at the Institute as an instructor in the department of chemistry, and then joined the staff of Ford, Bacon & Davis as an appraisal engineer. He then spent a year and a half as engineer and draftsman with the Differential Steel Car Co. of Findlay, Ohio, where he designed and supervised the construction of several types of automatic side-dump bodies for motor trucks and electric-railway cars.

C. M. Bonnell, Jr.

Mr. Bonnell became connected with the Freight Container Bureau of the American Railway Association in July, 1921, when work upon shipping containers was started. His initial training in this field was obtained under Colonel Dunn of the Bureau of Explosives, who supervised the work until Feb., 1925, and at the Forest Products Laboratory of the United States Department of Agriculture at the University of Wisconsin.

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F. H. Brown, author of Manufacture of Commercial Steel Helical Springs, has been closely associated with springs for thirty-three years. He entered the employment of the Washburn & Moen Manufacturing Co. in 1892 as foreman of the stockroom. He was advanced through the foremanship of the various divisions of spring manufacturing, and was promoted to his present position of spring engineer and superintendent of spring manufacture by the American Steel Wire Co. in 1901.

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F. D. Campbell

Frank D. Campbell, who presents a paper on Materials-Handling Problems and Their Solution, was engaged in experimental work in several lines of industrial-plant development, including improvement of processes and layout, from 1907 to 1916. In 1916 he became associated as engineer along these lines with a holding company for a large number of industries. Mr. Campbell joined the Lamson Company in 1918, as engineer upon the industrial application of their conveyor and pneumatic-tube equipment.

Since 1924 Mr. Campbell has been in business for himself as consulting engineer in materials handling, and has designed and superintended the installation of numerous large and successful systems which have been the first to be applied to their particular industries in several instances.

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F. E. Cardullo

Forrest E. Cardullo, author of The Question Mark in Machine Design, was born in Buffalo, N. Y., in 1879, and graduated at Cornell University with the degree of M.E. in 1901. From 1902 to 1905 he was draftsman and designer with the Holly Mfg. Co. of Buffalo, his work dealing with heavy gas engines and steam pumping engines. For three years following this he was instructor and assistant professor of mechanical-engineering subjects at Syracuse University. He became professor of mechanical engineering at New Hampshire State College in 1908, remaining there until 1914 when he accepted a similar position at the University of Texas.

Mr. Cardullo re-entered industrial work in 1915 as chief draftsman of the Pierce-Arrow Motor Car Co. In 1917 and 1918 he was chief engineer of tests for the Curtiss Aeroplane Co. He joined The G. A. Gray Co. of Cincinnati at the close of the war, and as their chief engineer has since made important improvements in planer design.

Mr. Cardullo has done much technical writing, covering the subjects of thermodynamics, industrial administration, and lately of machine-tool design. He is a member of the Special Committee on Cutting and Forming of Metals of the A.S.M.E.

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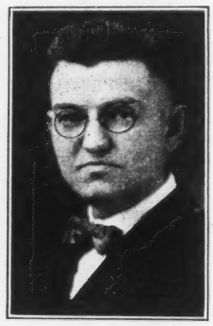

Ollison Craig

Ollison Craig, joint author with R. Sanford Riley of the paper on The Development of a Unit Pulverizer, was born at Sullivan, Ill., in 1888. He was graduated from the University of Illinois in 1909 with the degree of B.S. in Mechanical Engineering, and in 1920 received the degree of M.E. For seven years subsequent to his graduation he was engaged in engineering educational work at University of Colorado, at Vanderbilt University, and at Iowa State College.

In 1918 Mr. Craig joined the firm of Chas. L. Pillsbury Co., consulting engineers of

Minneapolis, remaining with them for two years. For the past five years he has been research engineer with the Under-Feed Stoker Co. of America and with the Riley Stoker Corp. He joined the A.S.M.E. in 1921.

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CHARLES FAIR

Charles Fair, author of High-Speed Induction Motors and Frequency Changers, has for the past six years been chief engineer for Baxter D. Whitney & Sons, Inc., manufacturers of woodworking machinery at Winchendon, Mass. Previous to this period he was for eighteen years with the General Electric Co. as engineer specializing in electric drives for machine tools, woodworking machinery, etc.

Mr. Fair, who for a number of years was a member of the A.S.M.E. Committee on Machine Shop Practice, has been the author of a number of papers on the application of motors to machine tools, standardization of machine-tool electrical equipment, reversing of high-speed motors, and kindred subjects.

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LILLIAN M. GILBRETH

Lillian M. Gilbreth, author of the paper on The Present State of Industrial Psychology, is president and treasurer of Frank B. Gilbreth, Inc. Since June 1924 she has carried on single-handed the consulting-engineering practice established by her husband, the late Frank Bunker Gilbreth, in the development of which she played an important part.

Mrs. Gilbreth was educated at the University of California where she received her B.S. degree in 1900 and M.L. degree in 1902. In 1915 she was given the degree of Ph.D. by Brown University. She is a member of Phi Beta Kappa.

In 1904 she married Mr. Gilbreth and for twenty years coöperated with him in his studies of motion and fatigue as affecting industrial work. Mrs. Gilbreth has done much writing, both alone and in coöperation with her husband, being the contributor of the article on Scientific Management in the New International Encyclopedia. At the same time Mrs. Gilbreth has brought up her remarkable family of eleven children.

Mrs. Gilbreth is a member of the American Psychological Association, the Taylor Society, the American Management Association, and the Acadamy Masaryk, and in April, 1925, was elected the second Honorary Member of the Society of Industrial Engineers.

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G. W. GREENWOOD

G. W. Greenwood, author of The Making of Special Slide Rules, entered Oxford University, England, after less than a year in a preparatory school, worked his way through and was graduated in 1900 with a First Class in their Honor School of Mathematics. He received a B.A. at that time and an M.A. degree four years later.

From 1900 to 1906 Mr. Greenwood held the chair of mathematics and astronomy at McKendree College. At the end of this time he returned to accounting, in which he had been engaged before entering Oxford. He was successively auditor, secretary and treasurer, and general manager of a corporation. At the present time Mr. Greenwood is specializing in accounting systems, and in addition to other executive positions which he holds, he is secretary and treasurer of the Warren Lumber Co. of Uniontown, Pa.

He has written upon many subjects and was lately the author of the chapter on mathematics in Management's Handbook. He is a fellow of the American Association for the Advancement of Science.

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A. JENSEN, JR.

Andrea Jensen, Jr., who contributes the paper on The Electric Molder, is industrial consultant with The Piqua Handle & Mfg. Co., at Marquette, Mich. He studied mechanical and electrical engineering at the Massachusetts Institute of Technology, and after his graduation became affiliated with the Cooley & Marvin Co. of Boston. His work with this concern was along the lines of industrial engineering, and led to his present position.

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Wilbur L. Jones, who writes upon Recent Advances in Methods of Glue Evaluation, was graduated from the Engineering School of New York University. He was then for several years engineering chemist for the City of New York. During the World War Mr. Jones was in charge of the inspection of glues for the U. S. Government, and eventually was made chairman of the Technical Committee on Glue of the Federal Specifications Board. At the present time Mr. Jones is conducting a laboratory of his own in Madison, Wis., where he specializes upon commercial glue testing, and upon research in the field of adhesives and wood compositions.

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H. F. KURTZ

Henry F. Kurtz, who presents a paper upon Principles and Advantages of Optical Methods for Measuring Machine Parts, is a member of the staff of the Scientific Bureau at the Bausch & Lomb Optical Company, Rochester, N. Y., where his attention is directed chiefly to the design of optical apparatus. He was graduated in 1906 from the Mechanics' Institute, Rochester.

After five years' service in various departments of the Bausch & Lomb plant, he entered upon four years of special training at the University of Rochester, with emphasis upon physics and mathematics. This was in preparation for the work of optical engineering in which he has since been engaged. He is a member of the Optical Society of America, of which he is a member of the committee on publications, and is a past-president of the Rochester section of that society.

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W. H. LARKIN, JR.

William H. Larkin, Jr., author of the paper on The Supply of Industrial Power, was graduated from the Worcester Polytechnic Institute in 1893 with the degree of B.S. and in 1903 received the degree of M.E. from this same institution. For several years immediately following his graduation he was master mechanic, U. S. Light House Establishment, 1st and 2nd Districts, under the régime of the Corps of Engineers, U. S. A.

In 1901 Mr. Larkin became chief mechanical engineer on the staff of the late Frank B. Gilbreth, who at that period was a general contractor with offices in Boston, New York, and London. In this connection Mr. Larkin was in 1903 the erecting engineer in charge of large-capacity machinery for concrete feeding, mixing, and placing at the Howden and the Derwent dam projects near Sheffield, England. This work led him to become interested in the pneumatic conveying of concrete, and with J. W. Buzzell he took out the basic patents upon this method.

In 1907 he took charge of the traction locomotive and separator departments of the M. Rumely Co. with whom he remained five years. He left them to become plant engineer at Chelsea, Mass., for the U. S.

Rubber Co. In 1919 he was made division power and safety engineer of their mechanical goods factories, in 1924 engineer of their steam plants, and in 1925 power engineer of their general division. Mr. Larkin, who joined the A.S.M.E. in 1904, has been a member of the Sub-Committee on Care of Power Boilers since its organization. He is a charter member of the Plant Engineers' Club of Boston.

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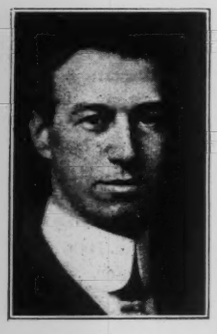

H. T. MOORE

Harold T. Moore, contributor of the paper on The Influence of Plant Design on Plant Efficiency, was graduated from the University of Pennsylvania in 1901 with the degree of B.S. in Mechanical Engineering, and was awarded his M.E. degree in 1904. His first position was that of draftsman with the Link Belt Co. of Philadelphia from 1901 to 1903. He then served as assistant industrial engineer with Dodge & Day of Philadelphia, for five years. He next served for two years as production engineer with the Bridgeport Brass Co. In 1910 Mr. Moore became assistant to the president of Cruse Kemper Co. at Ambler, Pa.

Since 1912 Mr. Moore has been chief industrial engineer of Day & Zimmerman in Philadelphia, with a wide range of duties covering both engineering and management subjects. He became an associate-member of the A.S.M.E. in 1907 and is also a member of the National Electric Light Association and of the Engineers' Club of Philadelphia.

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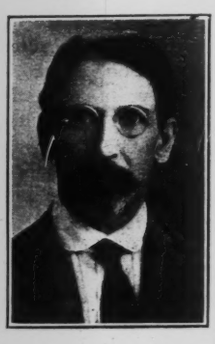

S. A. MOSS

Sanford A. Moss, author of Centrifugal Compressors for Diesel Engines, began his engineering gareer by serving an apprenticeship as a machinist in a shop in San Francisco, Cal. Upon the completion of this apprenticeship he entered the University of California, where he received a Bachelors' degree in 1896.

After his graduation he spent seven years as draftsman and engineer, and as an instructor at both the University of California and at Cornell. During this period be carried on post-graduate work, and in 1903 he received his Doctor's degree from Cornell University. Since that time he has been engaged in mechanical-engineering research work with the General Electric Co. He has been active in varied lines, including those of steam turbines, gas turbines, superchargers for internal-combustion engines, and centrifugal compressors. Dr. Moss has been a member of the A.S.M.E. since 1903, and has been the author of many papers and discussions which appear in the Transactions of the Society.

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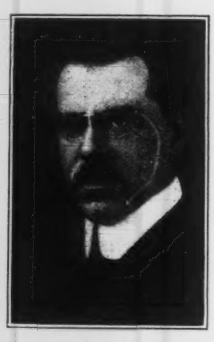

W. N. POLAKOV

Walter N. Polakov, author of Carbon Dioxide as an Index of Fatigue, was born at Luga, Russia, in 1879. He received his education at the Moscow Royal Institute of Technology, at Dresden (M.E.), and at the University of Moscow (Med.). He began his engineering work as assistant erecting foreman in the locomotive works at Tula, Russia, in 1902. From 1903 to 1906 he was chief engineer of the Department of Navigation to Harbors on the Caspian Sea.

Mr. Polakov became associated with the American Locomotive Co. as power-plant betterment engineer in 1907, remaining with them for two years. He next served for a year as expert consulting engineer on the Board of Estimate and Apportionment, City of New York. Following this he was associated with Harrington Emerson, H. L. Gantt, and Day & Zimmerman in installing scientific management in public utilities.

Mr. Polakov established himself as a consulting engineer in New York in 1915, specializing in management problems, and incorporated in 1919 as Walter N. Polakov & Co. Inc.

He has invented many systems used in control and management and has written several books. He is president of the Association of Russian Engineers in the U. S.

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A. A. RAYMOND

A. A. Raymond, who contributes the paper on Gas Engines of the Maryland Plant of the Bethlehem Steel Corp., has been superintendent of the gas-engine department of the Sparrows Point plant of this concern since 1923. He was first employed by them in 1913 at their plant in Bethlehem, Pa., leaving at the end of two years to become connected with the Allis-Chalmers Mfg. Co. He returned to the Bethlehem Steel Corp. in 1919 as a member of the staff of the power department.

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R. Sanford Riley, joint author with Ollison Craig of The Development of a Unit Pulverizer, was born in Hamilton, Ont., in 1874. He was educated in the schools of Winnipeg Manitoba, and at the Worcester Polytechnic Institute, where he was graduated with the degree of B.S. in 1896.

R. SANFORD RILEY

He was an instructor in mechanical engineering at M.I.T. for two years following his graduation, and then served for a time as a draftsman at Cramp's Shipyard. From 1898 to 1903 he served as marine engineer, spending much of this period at sea. He was then assistant to the chief engineer of the New York Shipbuilding Co., at Camden, N. J., leaving them in 1906 to become general manager of the American Ship Windlass Co. of Providence.

In 1911 Mr. Riley organized the Sanford-Riley Stoker Corp., at Worcester, Mass., of which he has since been president. He developed the Taylor stoker, and later invented the Riley stoker, and is the patentee of some thirty inventions. During the war Mr. Riley was organizer and head of the Performance Section of the Emergency Fleet Corp., and was in charge of all trial trips. He joined the A.S.M.E. in 1906.

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J. W. ROCKEFELLER

J. W. Rockefeller, Jr., who contributes the paper on Characteristics of Weighing Springs, was graduated in 1920 from the Massachusetts Institute of Technology with the degree of B.S. During his college days he was interested in writing. He joined the organization of John Chatillon & Sons, of New York, upon his graduation, serving first in the engineering service department. At the present time he is manager of the spring division of this company, having charge of both design and production.

When the Spring Research Committee of the A.S.M.E. was formed Mr. Rockefeller was appointed to represent the American Society for Steel Treating, and at the first meeting of this committee, held at Cleveland, in May, 1924, he was elected chairman. In addition to his regular work Mr. Rockefeller has served as instructor in metallography at the evening classes of the Brooklyn Polytechnic Institute, and has written many articles for the technical press.

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B. W. St. Clair, author of the paper on Springs for Electrical Measuring Instruments, was born in Louisville, Ky., in 1892. Upon graduation from the Louisville Male

B. W. ST. CLAIR

High School in 1910, he entered the standardizing laboratory of the General Electric Co. at Schenectady, N. Y., where he spent a year. At the end of that period he entered the meter department of the Louisville Lighting Co., for a year. He returned to the General Electric Co. in 1912, and was this time assigned to their standardizing laboratory in Lynn, Mass., where he has since been located with the exception of a period during and shortly after the war. That time Mr. St. Clair spent in research on submarine detector development, with the rank of ensign, U.S.R.F. Since his release from the naval service in 1919, Mr. St. Clair has held the position of engineer in charge of the standardization laboratory at Lynn.

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H. L. SEWARD

Herbert L. Seward, author of Graphical Methods of Calculation, was born at Guilford, Conn., in 1885, and was educated at Yale University, from which he received his Ph.B. degree in 1906 and M.E. degree in 1908. Since his graduation he has been continuously a member of the teaching staff of the Sheffield Scientific School at Yale, with the exception of the World War period. During that period he developed and operated the U. S. Navy Steam Engineering School at Stevens Institute, with the rank of lieutenant commander, U.S.N.R.F.

Professor Seward has engaged in practical and professional work every summer with such concerns as the Westinghouse Machine Co., the Westinghouse Electric & Mfg. Co., Link Belt Co., and Western Electric Co. He also spent one summer with the U. S. Shipping Board devising methods of fuel-oil conservation. He represents the A.S.M.E. on the Fuel Conservation Committee of the Shipping Board. He is now associate professor of mechanical engineering at the Sheffield Scientific School.

Professor Seward has done much technical writing along the lines of his present paper, including a book entitled Graphical Computations, and is an authority upon this subject. He has always been an ardent lover of the sea and holds a marine chief engineer's license, unlimited tonnage. He is staff navigator of the Connecticut Naval Militia, which he was active in organizing, and commands U. S. S. Eagle No. 27. He is the present chairman of the New Haven Section, A.S.M.E.

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W. TRINKS

W. Trinks, who presents in this issue the paper on Industrial Furnaces has been professor of mechanical engineering at the Carnegie Institute of Technology at Pittsburgh, since 1905. Professor Trinks was born in Berlin, Germany, in 1874. He studied mechanical engineering at Charlottenburg, from which institution he was graduated in 1897. He then began work with Schuchtermann & Kremer in Dortmund, Germany, and after coming to the United States he was first employed at Cramp's Shipyard. Following this he worked for a time at the Southwark Foundry in Philadelphia, and at the Westinghouse Machine Co. in Pittsburgh.

In 1902 he became chief mechanical engineer of the William Tod Co., of Youngstown, Ohio, where he remained until he took up his duties at the Carnegie Institute. In addition to teaching, Professor Trinks is consulting engineer for four widely known concerns, and is vice-president of a fifth. He has written many papers and several books, and is a member of the A.S.M.E., the S.A.E. and several other American and German technical societies.

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W. P. WOOD

W. P. Wood, who contributes in this issue the paper entitled Bolts for Use in Power-Plant Construction, is a graduate of the University of Michigan, from which he received the degree of A.B. in 1912, B.S. in chemical engineering in 1914, and M.S. in engineering in 1916. He was instructor in chemistry and assistant professor of chemical engineering at Michigan State College from 1914 to 1917. During the war he served as supervisor of materials, U. S. Bureau of Aircraft Production. In 1919 he took up his present duties as assistant professor of metallurgical engineering at the University of Michigan. Professor Wood is a member of the American Chemical Society and the American Society for Steel Treating, and has published several papers dealing with heat treatment of steel and the corrosion of iron and steel.